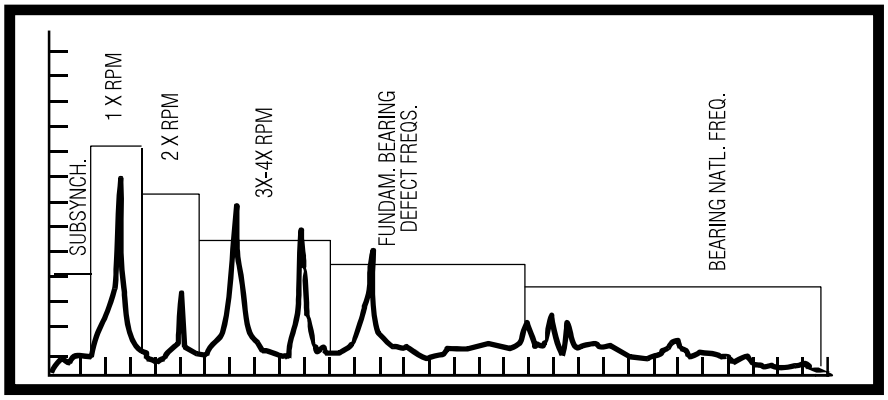
ANALYSIS I

How To Implement An Effective Condition Monitoring Program Using Vibration Analysis

METRIC VERSION



Authored By:
Mr. James E. Berry P.E.
of Technical Associates of Charlotte, Inc.

Technical Associates Of Charlotte, P.C.

347 North Caswell Road Charlotte, NC 28284, U.S.A. TELEPHONE: (704) 333-9011 FAX: (704) 333-1728

SPECIALISTS IN PREDICTIVE MAINTENANCE, MACHINERY DIAGNOSTICS, AND VIBRATION REDUCTION

PREDICTIVE MAINTENANCE AND VIBRATION SIGNATURE ANALYSIS I

By: James E. Berry, P.E., Mechanical Engineer

Technical Associates of Charlotte, P.C.
347 N. Caswell Rd.
Charlotte, NC 28204
(704) 333-9011

SPECIALISTS IN PREDICTIVE MAINTENANCE, MACHINERY DIAGNOSTICS, AND VIBRATION REDUCTION

We thank you for the opportunity to assist you in development of your vibration analysis and condition monitoring skills. As you can see from the length and breadth of the information within this text, many variables go into application of the material and so much depends upon the knowledge and judgement of the analyst and/or technician. The author and his company and /or companies can assume no responsibility for results in spite of many years of successful and trouble-free personal application.

First Edition: February, 1994
Second Edition: June, 1997
Third Edition: January, 2002

Copyright © 1990, 1991, 1992, 1993, 1994,1997 & 2002 Technical Associates of Charlotte, P.C. All rights reserved. This booklet, or parts thereof, may not be reproduced in any form without written permission from an officer of the company.

R-0102-1

TABLE OF CONTENTS AND SEMINAR AGENDA

PREDICTIVE MAINTENANCE AND VIBRATION SIGNATURE ANALYSIS I

SECTION	SUBJECT	PAGE
1.	**ANALYSIS I SEMINAR OVERVIEW**	**1-1**
2.	**WHAT IS VIBRATION AND HOW CAN IT BE USED TO EVALUATE MACHINERY CONDITION?**	**2-1**
	2.0 Introduction	2-1
	2.01 What is Vibration Frequency and How Does it Relate to a Time Waveform?	2-3
	2.02 What is Vibration Displacement?	2-3
	2.03 What is Vibration Velocity?	2-4
	2.04 What is Vibration Acceleration?	2-4
	2.05 What is Vibration Phase?	2-5
	2.1 What is a Vibration Spectrum (Also Called an "FFT" or "Signature")?	2-7
	2.2 Difference Between RMS, Peak and Peak-To-Peak Vibration Amplitude?	2-11
	2.3 When to Use Displacement, Velocity, or Acceleration?	2-13
	2.4 How Much is Too Much Vibration?	2-14
	2.5 Understanding a Vibration Spectrum	2-22
	2.51 Effect on Frequency Accuracy of #FFT Lines Used	2-22
	2.52 Effect on Frequency Accuracy of Frequency Span Used	2-24
	2.6 What is Overall Vibration (Digital and Analog Overall Level)?	2-33
	2.7 Understanding Phase and Its Applications	2-37
	2.71 Definition of Phase	2-37
	2.72 How to Make Phase Measurements	2-37
	2.73 Using Phase Analysis in Vibration Diagnostics	2-39
	2.731 Evaluating Axial Motion of a Bearing Housing to Reveal a Possible Cocked Bearing or Bent Shaft	2-39
	2.732 Phase Behavior Due to Unbalance	2-40
	2.733 Phase Behavior Due to Looseness/Weakness	2-42
	2.734 Phase Behavior Due to Misalignment	2-42
	2.735 Using Phase Analysis to Find the Operating Deflection Shape of a Machine and Its Support Frame	2-44
3.	**OVERVIEW OF THE STRENGTHS AND WEAKNESSES OF TYPICAL VIBRATION INSTRUMENTS**	**3-1**
	3.0 Introduction	3-1
	3.1 Instrument Comparisons	3-1
	3.2 General Capabilities of Each Vibration Instrument Type	3-5
	3.21 Overall Level Vibration Meters	3-5
	3.22 Swept-Filter Analyzers	3-6
	3.23 FFT Programmable Data Collectors	3-6
	3.24 Real-Time Spectrum Analyzers	3-7
	3.25 Instrument Quality Tape Recorders	3-8
4.	**OVERVIEW OF VARIOUS VIBRATION TRANSDUCERS AND HOW TO PROPERLY SELECT THEM**	**4-1**

© Copyright 1997 Technical Associates Of Charlotte, P.C.

SECTION **SUBJECT** **PAGE**

4.0 Introduction..... 4-1
4.1 Types of Vibration Transducers and Their Optimum Applications..... 4-1
4.11 Accelerometers..... 4-4
4.12 Velocity Pickups..... 4-9
4.13 Non-Contact Eddy Current Displacement Probes..... 4-14
4.14 Shaft Contact Displacement Probes..... 4-18
4.141 Shaft Sticks..... 4-18
4.142 Shaft Riders..... 4-20
4.2 Selection Criteria For Transducers..... 4-21
4.3 Mounting of Transducers..... 4-23
4.31 Transducer Mounting Applications..... 4-23
Appendix - Specifications for Various Transducers from a Variety of Manufacturers..... 4-26

5. ROLE OF SPIKE ENERGY, HFD AND SHOCK PULSE (SPM) AND SPECIFICATION OF THEIR ALARM LEVELS AT VARIOUS SPEEDS..... 5-1

A. Spike Energy and Shock Pulse..... 5-1

6. USE OF VIBRATION SIGNATURE ANALYSIS TO DIAGNOSE MACHINE PROBLEMS..... 6-1

6.0 Use of Vibration Signature Analysis..... 6-1
Overview of 5-Page "Illustrated Vibration Diagnostic Chart"..... 6-4

6.01 Mass Unbalance..... 6-12
6.011 Force Unbalance..... 6-15
6.012 Couple Unbalance..... 6-15
6.013 Dynamic Unbalance..... 6-16
6.014 Overhung Rotor Unbalance..... 6-17
6.0141 Summary of Procedures for Balancing Overhung Rotors..... 6-17
6.015 Allowable Residual Unbalance and ISO Balance Quality Grade..... 6-21
6.02 Eccentric Rotors..... 6-27
6.03 Bent Shaft..... 6-30
6.04 Misalignment..... 6-32
6.041 Angular Misalignment..... 6-35
6.042 Parallel Misalignment..... 6-36
6.043 Misaligned Bearing Cocked on the Shaft..... 6-37
6.044 Coupling Problems..... 6-37
6.05 Natural Frequencies and Resonance..... 6-39
6.051 Natural Frequency..... 6-39
6.052 Resonance..... 6-42
6.06 Mechanical Looseness..... 6-43
6.061 Type A - Structural Frame/Base Looseness (1X RPM)..... 6-43
6.062 Type B - Looseness Due to Rocking Motion or Cracked Structure/Bearing Pedestal (2X RPM)..... 6-46
Type C - Loose Bearing in Housing or Improper Fit Between Component Parts (Multiple Harmonics Due to Nonlinearity Often Induced By Impulse Events)..... 6-48
6.07 Tracking of Rolling Element Bearing Failure Stages Using Vibration and High Frequency Enveloping and Demodulated Spectral Techniques..... 6-52
6.071 Optimum Vibration Parameter For Rolling Element Bearing Condition Evaluation (Acceleration, Velocity or Displacement)..... 6-55
6.072 Types of Vibration Spectra Caused by Defective Rolling Bearings..... 6-57

© Copyright 1997 Technical Associates Of Charlotte, P.C.

6.073 Typical Spectra for Tracking Failure Stages Through Which Rolling Element Bearings Pass.... 6-70
6.08 Introduction to Gear Problem Detection.... 6-80
6.081 Specification of Spectral Setup for Detecting Gear Wear.... 6-80
6.082 Indications of Gear Tooth Wear.... 6-82
6.09 Introduction to Electrical Problem Detection.... 6-86
6.091 Why Many Electrical Problems Occur at 2X Line Frequency.... 6-86
6.092 Stator Problems.... 6-87
6.093 Eccentric Rotor and Variable Air Gap.... 6-91
6.094 Rotor Problems.... 6-93
6.095 Important Closing Comments on Electrical Measurements.... 6-95
6.10 Belt Drive Problems.... 6-99
6.101 Worn, Loose or Mismatched Belts.... 6-100
6.102 Belt/Sheave Misalignment.... 6-102
6.103 Eccentric Sheaves.... 6-102
6.104 Belt Resonance.... 6-103
6.105 Excessive Motor Vibration at Fan Speed Due to Motor Frame/Foundation Resonances.... 6-103
6.106 Loose Pulley or Fan Hub.... 6-103

7. PROVEN METHOD FOR SPECIFYING SPECTRAL BAND ALARM LEVELS AND FREQUENCIES USING TODAY'S PREDICTIVE MAINTENANCE SOFTWARE SYSTEMS.... 7-1

7.0 Abstract.... 7-1
7.1 Introduction to Specifying Spectral Alarm Bands & Frequency Ranges.... 7-2
7.11 Two Types of Spectral Alarm Bands.... 7-3
7.12 Which Vibration Parameter to Use in Spectral Alarm Bands - Displacement, Velocity or Acceleration?.... 7-4
7.13 Review of Problems Detectable by Vibration Analysis.... 7-5
7.14 Specification of Overall Vibration alarm Levels and Explanation of the Origin of Table II "Overall Condition Rating" Chart.... 7-13
7.15 Specification of Spectral Alarm Levels and Frequency Bands Using Table III.... 7-16
Case A - General Rolling Element Bearing Machine Without Rotating Vanes: (Motors, Spindles, Gearbox Lower Frequency Measurements, etc.).... 7-17
Case B - General Sleeve Bearing Machine Without Rotating Vanes: (Sleeve Bearing Motors, Gearbox Lower Frequency Measurements, etc.).... 7-18
Case C - Gearbox High Frequency Points with Known Number of Teeth.... 7-19
Case D - Gearbox High Frequency Points with Unknown Number of Teeth.... 7-20
Case E - Motor Electrical Rotor Bar Pass Frequency Point (Single Point Usually Taken on Outboard Motor Bearing).... 7-21
Case F - Motor Electrical 12,000 CPM Measurement Point (Single Point Usually Taken on Inboard Motor Bearing).... 7-21
Case G - Special Machine Types.... 7-22
Type 1 - Centrifugal Machines with Known Number of Vanes (or Blades) and Rolling Element Bearings.... 7-22
Type 2 - Centrifugal Machines with Unknown Number of Vanes (or Blades) and Rolling Element Bearings.... 7-22
Type 3 - Centrifugal Machines with Known Number of Vanes (or Blades) and Sleeve Bearings.... 7-23
Type 4 - Centrifugal Machines with Unknown Number of Vanes (or Blades) and Sleeve Bearings.... 7-23
7.151 Examples - Specification of Spectral Alarm Bands for Sample Machines.... 7-23

© Copyright 1997 Technical Associates Of Charlotte, P.C.

SECTION **SUBJECT** **PAGE**

7.16 Periodic Reevaluation of Spectral alarm Band Setups on Each Family of Machines....7-29
7.161 Procedure for Evaluating the Effectiveness of Specified Overall Alarm Levels and Spectral Bands.... 7-31
7.162 Example - "Statistical Analysis of Overall Vibration Velocity in 4 Client Power Plants Using the Procedure Recommended Above"....7-32
7.17 Conclusions.... 7-34

8. COMMON PITFALLS IN EVERYDAY VIBRATION MEASUREMENTS....8-1

8.0 Introduction.... 8-1
8.1 General Consideration for Obtaining Consistent Quality Data.... 8-1
8.11 Choosing Measurement Locations.... 8-1
8.12 Machine and Point Identification.... 8-3
8.13 Measurement Parameters.... 8-7
8.14 Instrument Selection, Setup, and Condition.... 8-12
8.15 Measurement Techniques.... 8-14
8.16 Transducer Mounting and Probes....8-15
8.2 Effect of Transducer Mounting on Vibration Measurements....8-19

9. SETUP AND IMPLEMENTATION OF PREDICTIVE MAINTENANCE AND CONDITION MONITORING PROGRAMS....9-1

- "Flow Chart of Recommended Predictive Maintenance Programs".... 9-7
- "Sample PMP Machine Plant Layout".... 9-8
- "Sample Machinery Drawing and Configuration Chart".... 9-9
- "Criteria for Overall Condition Rating (Peak Overall Velocity, in/sec)".... 9-10
- "Sample Overall Condition Rating Report".... 9-12
- "Sample Overall Condition Rating Bar Graph".... 9-14
- "Sample Rank-Ordered Results and Recommendations Report"....9-16
- "Sample PMP Machine Repair Log".... 9-17
- "Latest Overall Measurement Report".... 9-19
- "Overall Alarm Exception Report".... 9-21
- "Spectral Band Alarm Report".... 9-23
- "Spectral Narrowband Envelope Alarm Report".... 9-34
- "Current Inspection Code Report".... 9-44
- "Trends and Waterfall Plots"....9-45
- "Effect of Vibration Acceptance Testing".... 9-49

10. REAL-WORLD CASE HISTORIES.... 10-1

(See Individual Table of Contents At Beginning of This Section)

© Copyright 1997 Technical Associates Of Charlotte, P.C.

SAMPLE

TECHNICAL ASSOCIATES PRE-SEMINAR QUESTIONNAIRE

SAMPLE

YOUR NAME: JOHN W. DOE
TITLE: Technician
COMPANY: XYZ Chemicals Inc.
LOCATION: Americana, TX
PHONE: (512) 222-1233 EXT: 765
EMAIL: _____ FAX: _____

SEMINAR DATES: April 8-11, 1997
SEMINAR LOCATION: Myrtle Beach, SC
SEMINAR LEVEL (CIRCLE) ENTRY (I) II III IVD ADV
VIBRATION EXPERIENCE: YRS. ____ MOS. 6
SPECTRAL ANAL. EXPER: YRS. ____ MOS. 0

PLEASE COMPLETE THE FOLLOWING QUESTIONNAIRE, THIS INFORMATION IS FOR USE IN THIS CLASS AND TO HELP US PLAN FUTURE SEMINARS: ***(SEE SAMPLE PRE-SEMINAR SURVEY ENCLOSED)***

A. TYPE OF INDUSTRY AND PRODUCT MANUFACTURED/SERVICES RENDERED BY YOUR COMPANY:

AUTOMOTIVE		CHEMICAL	✓	CONSUMER PROD.	
CONSULTANT		FOOD/BEVERAGES		FABRICA. METAL	
FIBERGLASS		HVAC		INSTRUM./S'WARE	
MINING		PETROLEUM		PHARMACEUTICAL	
PULP & PAPER		REPAIR		SYNTHETIC FIBER	
TEXTILES		TOBACCO		UTILITIES	
VIB. SERVICES		TRAINING SERVICES		OTHER (SPECIFY)	

B. COMPANY INFORMATION:

PRIMARY PRODUCT(S) AT YOUR FACILITY? Industrial Chemicals
NUMBER OF EMPLOYEES AT YOUR FACILITY? 525
DO YOU HAVE AN ONGOING VIBRATION/PMP PROGRAM? YES ✓ NO ____
IF YES, HOW LONG? 7 years
PERSON RESPONSIBLE FOR PROGRAM? NAME: J.P. DenHartog
TITLE: Plant mgr.

C. WHAT TYPE OF VIBRATION INSTRUMENTS ARE USED AT YOUR PLANT? (ENTER BOTH MANUFACTURER MODEL NO. AND QUANTITY IN PARENTHESES)

- OVERALL LEVEL METER (i.e, SKF Picalog): SKF Picalog (3)
- ANALYZER/BALANCER (i.e., IRD 885): IRD 885 (2)
- DATA COLLECTOR (i.e., IRD dataPAC 1500): CSI 2115 (4), CSI 2120 (2)
- 1 CHANNEL RTA (i.e., ROCKLAND 5815A): ____
- MULTI-CHANNEL DATA COLLECTOR (i.e., CSI 2120): CSI 2400 (1)
- 2 CHANNEL RTA (i.e., HP 35670): ____
- SWEPT-FILTER ANALYZER (i.e., SCHENCK VIBRAPORT 30): IRD 880 (1)
- MULTI-CHANNEL RTA (i.e., SA 390): SA 390 (1)

D. WHAT IS THE BRAND NAME AND MODEL # OF THE PRIMARY DATA COLLECTOR USED AT YOUR PLANT? *(INDICATE MODEL NO. IN BLANK)*

B&K ____ CSI 2120 DI ____ DLI ____ ENTEK IRD ____
SKF ____ TEC ____ VITEC ____ OTHER ____

E. WHAT PMP SOFTWARE AND VERSION # DO YOU USE? (PLACE CHECK MARK BESIDE ALL THAT YOU NOW USE)

7090 ____
MASTERTREND (DOS) ____
IQ 2000 ____
DLI Advanced ALERT ____
MASTERTREND (WIN) Ver 3.06
PRISM² (DOS) ____
PRISM⁴ (WIN) ____
EMONITOR (DOS) ____
EMONITOR (WIN) ____
OTHER ____

F. EXPERT DIAGNOSTIC SOFTWARE USED AT YOUR PLANT: (PLACE CHECK MARK)

AMETHYST	____	CHARLIE	____	EXAMINE	____	EXPLORE	____
EXPERTALERT	____	MDU	____	MOTORCHECK	____		
MOTORMONITOR	____	NSPECTR	✓	PRISM^4 PRO	____	ROMEX	____
VARMINT	____						
OTHER (PLEASE IDENTIFY)	____					NONE	____

G. EXPERIMENTAL MODAL ANALYSIS SOFTWARE USED AT YOUR PLANT (CHECK ALL THOSE WHICH YOU NOW USE).

LMS ____ MESCOPE ____ SDRC ____ STAR ✓ ZONIC ____ OTHER (SPECIF Y) ____

H. HAVE YOU PERFORMED FINITE ELEMENT MODELLING? IF SO, WITH WHAT SOFTWARE?

YES ____ NO ✓ SOFTWARE USED? ____

I. WHAT OTHER VIBRATION/PREDICTIVE MAINTENANCE SEMINARS HAVE YOU ATTENDED?

SEMINAR NAME	SEMINAR COMPANY	DATE ATTENDED (APPROXIMATE)
ENTRY	TECHNICAL ASSOC.	Jan. 1997

J. IS THERE ANY SPECIAL MACHINE TYPE IN YOUR FACILITY ON WHICH YOU NEED HELP ? AND/OR WOULD LIKE ADDITIONAL INFORMATION?

MACHINE NAME	BEARING TYPE	RPM	HP	DRIVE TYPE

K. PLEASE LIST ANY SPECIAL QUESTIONS OR TOPICS THAT YOU WOULD LIKE COVERED DURING THIS SEMINAR?

1) How can you accurately diagnose Rolling Element Brg problems and estimate their remaining life?

2) Is there any technique now available to detect problems on an intermediate gear inside a large multi-stage Gearbox?

L. HOW DID YOU FIND OUT ABOUT THIS SEMINAR? (PLEASE CHECK PRIMARY CONTACT)

BROCHURE	✓	DIRECT SALES	____	PHONE CALL	____
COLLEAGUE	____	FORMER ATTENDER	✓		
MAGAZINE AD (If So, Which Magazine)?	____				
OTHER (PLEASE INDICATE SOURCE)	____				

M. WHAT PROMPTED YOU TO ATTEND THIS PARTICULAR SEMINAR? (PLEASE CHECK ALL THAT APPLY.)

ADVANCEMENT	____	SEMINAR LOCATION	____	SPEAKER	____
SUBJECT MATTER	____	WAS RECOMMENDED	✓	OTHER	____

PLEASE SUBMIT THIS COMPLETED FORM AND MAIL/FAX TO TECHNICAL ASSOCIATES BEFORE ATTENDING SEMINAR. THANK YOU. WE APPRECIATE YOUR TAKING THE TIME TO HELP US CUSTOMIZE AND ENHANCE THIS SEMINAR TO ENSURE WE MEET YOUR PARTICULAR NEEDS.

TECHNICAL ASSOCIATES OF CHARLOTTE, P.C.

TECHNICAL ASSOCIATES PRE-SEMINAR QUESTIONNAIRE

YOUR NAME: _____ SEMINAR DATES: _____
TITLE: _____ SEMINAR LOCATION: _____
COMPANY: _____ SEMINAR LEVEL (CIRCLE) ENTRY I II III ADV
LOCATION: _____ VIBRATION EXPERIENCE: YRS. ____ MOS. _____
PHONE: _____ EXT: _____ SPECTRAL ANAL. EXPER: YRS. ____ MOS. _____
EMAIL: _____ FAX: _____

PLEASE COMPLETE THE FOLLOWING QUESTIONNAIRE, THIS INFORMATION IS FOR USE IN THIS CLASS AND TO HELP US PLAN FUTURE SEMINARS: ***(SEE SAMPLE PRE-SEMINAR SURVEY ENCLOSED)***

A. TYPE OF INDUSTRY AND PRODUCT MANUFACTURED/SERVICES RENDERED BY YOUR COMPANY:

AUTOMOTIVE	_____	CHEMICAL	_____	CONSUMER PROD.	_____
CONSULTANT	_____	FOOD/BEVERAGES	_____	FABRICA. METAL	_____
FIBERGLASS	_____	HVAC	_____	INSTRUM./S'WARE	_____
MINING	_____	PETROLEUM	_____	PHARMACEUTICAL	_____
PULP & PAPER	_____	REPAIR	_____	SYNTHETIC FIBER	_____
TEXTILES	_____	TOBACCO	_____	UTILITIES	_____
VIB. SERVICES	_____	TRAINING SERVICES	_____	OTHER (SPECIFY)	_____

B. COMPANY INFORMATION:

PRIMARY PRODUCT(S) AT YOUR FACILITY? _____
NUMBER OF EMPLOYEES AT YOUR FACILITY? _____
DO YOU HAVE AN ONGOING VIBRATION/PMP PROGRAM? YES _____ NO _____
IF YES, HOW LONG? _____
PERSON RESPONSIBLE FOR PROGRAM? NAME: _____
TITLE: _____

C. WHAT TYPE OF VIBRATION INSTRUMENTS ARE USED AT YOUR PLANT? (ENTER BOTH MANUFACTURER MODEL NO. AND QUANTITY IN PARENTHESES)

- OVERALL LEVEL METER (i.e, SKF Picalog) _____
- DATA COLLECTOR (i.e., ENTEK DataPAC 1500) _____
- MULTI-CHANNEL DATA COLLECTOR (i.e., CSI 2120) _____
- SWEPT-FILTER ANALYZER (i.e., SCHENCK VIBRAPORT 30) _____
- ANALYZER/BALANCER (i.e., IRD 885) _____
- 1 CHANNEL RTA (i.e., ROCKLAND 5815A) _____
- 2 CHANNEL RTA (i.e., HP 35670) _____
- MULTI-CHANNEL RTA (i.e., SA 390) _____

D. WHAT IS THE BRAND NAME AND MODEL # OF THE PRIMARY DATA COLLECTOR USED AT YOUR PLANT? *(INDICATE MODEL NO. IN BLANK)*

B&K _____ COMMTEST _____ CSI _____ DI _____ DLI _____
ENTEK IRD _____ SKF _____ TEC _____ VITEC _____ OTHER _____

E. WHAT PMP SOFTWARE AND VERSION # DO YOU USE? (PLACE CHECK MARK BESIDE ALL THAT YOU NOW USE)

ALERT _____	EMONITOR (DOS) _____	IQ 2000 _____
MAS _____	MASTERTREND _____	ODYSSEY _____
PRISM[4] _____	RBMWARE _____	OTHER _____

F. EXPERT DIAGNOSTIC SOFTWARE USED AT YOUR PLANT: (PLACE CHECK MARK)

AMETHYST _____ CHARLIE _____ EXAMINE _____ EXPLORE _____
EXPERTALERT _____ MDU _____ MOTORCHECK _____
MOTORMONITOR _____ NSPECTR _____ PRISM^4 PRO _____ ROMEX _____
VARMINT _____
OTHER (PLEASE IDENTIFY) _____ NONE _____

G. EXPERIMENTAL MODAL ANALYSIS SOFTWARE USED AT YOUR PLANT (CHECK ALL THOSE WHICH YOU NOW USE).

LMS ____ MESCOPE ____ SDRC ____ STAR ____ ZONIC ____ OTHER (SPECIFY) ____

H. HAVE YOU PERFORMED FINITE ELEMENT MODELLING? IF SO, WITH WHAT SOFTWARE?

YES_____ NO_____ SOFTWARE USED?_____

I. WHAT OTHER VIBRATION/PREDICTIVE MAINTENANCE SEMINARS HAVE YOU ATTENDED?

SEMINAR NAME	SEMINAR COMPANY	DATE ATTENDED (APPROXIMATE)

J. IS THERE ANY SPECIAL MACHINE TYPE IN YOUR FACILITY ON WHICH YOU NEED HELP ? AND/OR WOULD LIKE ADDITIONAL INFORMATION?

MACHINE NAME	BEARING TYPE	RPM	HP	DRIVE TYPE

K. PLEASE LIST ANY SPECIAL QUESTIONS OR TOPICS THAT YOU WOULD LIKE COVERED DURING THIS SEMINAR?

_____
_____
_____
_____
_____
_____
_____

L. HOW DID YOU FIND OUT ABOUT THIS SEMINAR? (PLEASE CHECK PRIMARY CONTACT)

BROCHURE _____ DIRECT SALES _____ PHONE CALL _____
COLLEAGUE _____ FORMER ATTENDER _____
MAGAZINE AD (If So, Which Magazine)? _____
OTHER (PLEASE INDICATE SOURCE) _____

M. WHAT PROMPTED YOU TO ATTEND THIS PARTICULAR SEMINAR? (PLEASE CHECK ALL THAT APPLY.)

ADVANCEMENT _____ SEMINAR LOCATION _____ SPEAKER _____
SUBJECT MATTER _____ WAS RECOMMENDED _____ OTHER _____

PLEASE SUBMIT THIS COMPLETED FORM AND MAIL/FAX TO TECHNICAL ASSOCIATES BEFORE ATTENDING SEMINAR. THANK YOU. WE APPRECIATE YOUR TAKING THE TIME TO HELP US CUSTOMIZE AND ENHANCE THIS SEMINAR TO ENSURE WE MEET YOUR PARTICULAR NEEDS.

TECHNICAL ASSOCIATES OF CHARLOTTE, P.C.

POST-SEMINAR EVALUATION

SEMINAR DATES: _____ SEMINAR LEVEL (CIRCLE): ENTRY I II III ADV.

ATTENDEE NAME: _____ TITLE: _____

COMPANY NAME: _____ YOUR VIBRATION EXPERIENCE: YEARS _____ MONTHS _____

COMPANY LOCATION: _____ YOUR SPECTRAL ANAL. EXPER: YEARS _____ MONTHS _____

SEMINAR LOCATION: _____

INSTRUCTOR: _____

EVALUATION TOPICS	LOW									HIGH
RATING >	1	2	3	4	5	6	7	8	9	10
1. EVALUATE CONTENT OF THIS SEMINAR.										
2. EVALUATE PERFORMANCE OF INSTRUCTOR:										
A. KNOWLEDGE OF SUBJECT										
B. PRESENTATION CAPABILITIES										
C. ENTHUSIASM DISPLAYED										
D. OVERALL PERFORMANCE RATING										
3. HOW APPLICABLE TO YOUR JOB HAS BEEN THE CONTENT OF THIS SEMINAR AND SEMINAR TEXT?										
4. CAN WHAT YOU HAVE LEARNED MAKE YOU MORE EFFECTIVE IN YOUR JOB?										
5. HOW WOULD YOU RATE THE SEMINAR FACILITIES:										
A. MEETING ROOM -										
B. SLEEPING ROOM -										
C. FOOD SERVICE -										
6. ARE YOU INTERESTED IN FURTHER TRAINING IN THIS AREA OF VIBRATION/PREDICTIVE MAINTENANCE?										

CONCENTRATION ON SEMINAR:

1. ON WHAT SHOULD *MORE* TIME BE CONCENTRATED? _____

2. ON WHAT SHOULD *LESS* TIME BE CONCENTRATED? _____

3. WHAT WOULD YOU SUGGEST TO *IMPROVE* FUTURE SEMINARS? _____

4. DO YOU HAVE ANY *ADDITIONAL COMMENTS* ABOUT THIS SEMINAR, THE INSTRUCTORS, THE COMPANY OR THE FACILITIES?

WE WANT TO GENUINELY THANK YOU FOR THE PRIVILEGE OF SHARING WITH YOU THIS WEEK.
WE HOPE TO SEE YOU AGAIN IN THE FUTURE.

RECOMMENDED PERIODICALS FOR THOSE INTERESTED IN PREDICTIVE MAINTENANCE

1. *Sound and Vibration Magazine*

P.O. Box 40416
Bay Village, OH 44140
Mr. Jack Mowry, Editor and Publisher
Phone: 216-835-0101
Fax : 216-835-9303

Terms: Normally free for bona fide qualified personnel concentrating in the Sound and Vibration Analysis/Plant Engineering Technologies. Non-qualified personnel - $25/per year within the U.S.

Comments: This is a monthly publication that normally will include approximately 4-6 issues per year devoted to Predictive Maintenance. Their Predictive Maintenance articles are usually practical and in good depth; normally contain real "meat" for the PPM vibration analyst. *Sound and Vibration* has been published for over 25 years.

2. *Vibrations Magazine*

The Vibration Institute
6262 South Kingery Hwy, Suite 212
Willowbrook, IL 60514
Institute Director - Dr. Ronald Eshleman
Phone: 630-654-2254
Fax : 630-654-2271

Terms: *Vibrations Magazine* is sent to Vibration Institute members as part of their annual fee, (approx. $45 per year). It is available for subscription to non-members at $55/per year; $60/foreign.

This is a quarterly publication of the Vibration Institute. Always contains very practical and useful Predictive Maintenance Articles and Case Histories. Well worth the small investment.

Comments: Yearly Vibration Institute fee includes reduced proceedings for that year if desired for the National Conference normally held in June. They normally meet once per year at a fee of about $675/per person, ($600/person for Institute members) including conference proceedings notes and mini-seminar papers. All of the papers presented, as well as mini-courses, at the meeting are filled with "meat" for the Predictive Maintenance Vibration Analyst. *Vibrations Magazine* was first published in 1985 although the Institute has been in existence since approximately 1972, with their first annual meeting in 1977. The Vibration Institute has several chapters located around the United States which normally meet on a quarterly basis. The Carolinas' Vibration Institute Chapter normally meets in Greenville, SC; Charleston, SC; Columbia, SC; Charlotte, NC; Raleigh, NC; and in the Winston Salem, NC areas. For Institute membership information, please contact: Dr. Ron Eshleman at 630-654-2254. When doing so, be sure to ask what regional chapter is located to your area. Membership fees for the "Annual Meeting Proceedings" are $30/per year (normal cost is approx. $60/per year for proceedings if annual meeting is not attended). Please tell Ron that we recommended you joining the Vibration Institute when you call or write to him.

3. *P/PM Technology Magazine*

P.O. Box 1706
Minden, NV 89423-1706 (Pacific Coast Time)
Phone: 702-267-3970; 800-848-8324
Fax : 702-267-3941
Publisher- Mr. Ronald James; Assistant Publisher: Susan Estes

Terms: $42/per year for qualified USA subscribers, (individuals and establishments involved with industrial plant and facilities maintenance; subscribers must be associated in engineering, maintenance, purchasing or management capacity). $60/year for unqualified subscribers.

Comments: This is a bi-monthly magazine with articles about all facets of PPM Technologies, including Vibration Analysis, Oil Analysis, Infrared Thermography, Ultrasonics, Steam Trap Monitoring, Motor Current Signature Analysis, etc. These are normally good practical articles. Also includes some cost savings information, although does not necessarily include how these cost savings were truly determined. *P/PM Technology* also hosts at least one major conference per year in various parts of the United States. Intensive training courses in a variety of condition monitoring technologies will also be offered in vibration analysis, root cause failure analysis, oil analysis, thermographic analysis, ultrasonic analysis, etc..)

4. *Maintenance Technology Magazine*

1209 Dundee Ave., Suite 8
Elgin, IL 60120
Phone: 800-554-7470
Fax : 804-304-8603
Publisher: Arthur L. Rice

Terms: $95/per year for non-qualified people This is a monthly magazine that usually has at least one article relating to Predictive Maintenance using vibration analysis within each issue. In addition to vibration, it likewise always offers other articles covering the many other technologies now within Predictive Maintenance.

5. Reliability Magazine

PO Box 856
Monteagle, TN 37356
Phone: 423-592-4848
Fax : 423-592-4849

Editor: Mr. Joseph L. Petersen

Terms: $49 per year in USA; $73 per year outside USA.

Comments: This bi-monthly magazine covers a wide variety of Condition Monitoring Technologies including Vibration Analysis, Training, Alignment, Infrared Thermography, Balancing, Lubrication Testing, CMMS and a unique category they entitle "Management Focus".

NOTE: In addition to these periodicals, many of the major predictive maintenance hardware and software vendors put out periodic newsletters. Some of these in fact do include some “real meat” in addition to their sales propaganda. We would recommend that you contact, particularly the vendor supplying your predictive maintenance system for their newsletter. Their newsletter will likewise advise you of updates in their current products.

TECHNICAL ASSOCIATES OF CHARLOTTE, P.C.

SINCE 1961

347 N. Caswell Road Charlotte, NC 28204 Ph: (704) 333-9011 Fax: (704) 333-1728 www.technicalassociates.net

PUBLICATION ORDER FORM

	QTY.	PRICE	TOTAL
ILLUSTRATED VIBRATION DIAGNOSTIC WALL CHART			
DWC-4 (LAMINATED 38" X 52")		$ 89.00	
USE OF VIBRATION SIGNATURE ANALYSIS TO DIAGNOSE MACHINE PROBLEMS (A COMPLETE COVERAGE FOR ALL PROBLEMS SHOWN ON THE **"ILLUSTRATED VIBRATION DIAGNOSTIC WALL CHART"**) NEW			
DWD-4 (190 PAGE BOOKLET)		$ 189.00	
BOTH THE **"ILLUSTRATED VIBRATION DIAGNOSTIC WALL CHART"** AND COMPLETE BOOKLET			
DWE-4		$ 249.00	
INTERACTIVE VIBRATION DAIGNOSTIC CHART - ELECTRONIC VERSION OF THE **"ILLUSTRATED VIBRATION DIAGNOSTIC WALL CHART"** NEW			
IVDC-1 (CD ROM)		$ 159.00	
PROVEN METHODS FOR SPECIFYING BOTH 6 SPECTRAL BANDS AS WELL AS NARROWBAND ALARM ENVELOPES USING TODAY'S PREDICTIVE MAINTENANCE SOFTWARE SYSTEMS - 3RD EDITION			
SAB-3 (50 PAGE BOOKLET ONLY)		$ 69.00	
VIBRATION DIAGNOSTICS TRAINING VIDEOTAPE SERIES			
VAT-8 (SET OF 7)		$ 2,495.00	
VIDEO WORKBOOKS			
VWB-8 (SET OF 7)		$ 200.00	
ADVANCED VIBRATION ANALYSIS - AC INDUCTION MOTORS TRAINING VIDEOTAPE SERIES			
AVT-3 (SET OF 2)		$ 995.00	
VIDEO WORKBOOKS **ATB-1**		$ 29.00	
BandAID SOFTWARE IS THE QUICKEST WAY TO DISPLAY INFORMATION REQUIRED FOR EACH MACHINE POINT WHEN BUILDING A DATEBASE			
BFW-3 (50 PAGE BOOKLET AND *WINDOWS* SOFTWARE)		$ 450.00	
BAS-2 (*DOS*)		$ 350.00	
VIBRATION ACCEPTANCE STANDARDS USING BandAID NEW			
VAS-1 (60 PAGE BOOKLET)		$ 89.00	
HOW TO AND HOW NOT TO CREATE MEANINGFUL NARROWBAND SPECTRAL ALARM ENVELOPES			
SAE-1 (53 PAGE BOOKLET)		$ 69.00	
TRACKING OF ROLLING ELEMENT BEARING FAILURE STAGES USING VIBRATION SIGNATURE ANALYSIS			
REB-1 (90 PAGE BOOKLET)		$ 69.00	
REQUIRED VIBRATION ANALYSIS TECHNIQUES AND INSTRUMENTATION ON LOW-SPEED MACHINES (PARTICULARLY 10 TO 300 RPM MACHINERY)			
LSM-1 (95 PAGE BOOKLET)		$ 69.00	
REQUIRED VIBRATION ANALYSIS TECHNIQUES AND INSTRUMENTATION ON HIGH-SPEED MACHINES (AT FREQUENCIES > 10,000 Hz OR 600,000 CPM)			
HSM-1 (29 PAGE BOOKLET)		$ 49.00	
HOW TO ANALYZE VARIABLE-SPEED MACHINES AND INCLUDE THEM IN A PREDICTIVE MAINTENANCE PROGRAM			
VSM-1 (20 PAGE BOOKLET)		$ 49.00	
HIGH FREQUENCY ENVELOPING AND DEMODULATION TECHNIQUES - 3 YEAR STUDY			
HFE-1 (60 PAGE BOOKLET)		$ 49.00	
HOW TIME WAVEFORM ANALYSIS AND TRUE SYNCHRONOUS TIME AVERAGING CAN BE USED TO EFFECTIVELY EVALUATE MACHINES WHICH HAVE PROBLEMS NOT NECESSARILY DETECTABLE BY FFT ANALYSIS ALONE			
TWA-1 (38 PAGE BOOKLET)		$ 49.00	
DIAGNOSTIC TESTS AND INSTRUMENTATION REQUIRED FOR NATURAL FREQUENCY PROBLEM DETECTION AND CORRECTION			
NFT-1 (60 PAGE BOOKLET)		$ 69.00	
COMPARISON OF MOTOR CURRENT ANALYSIS AND VIBRATION ANALYSIS IN DETECTING ROTOR BAR AND AIR GAP ECCENTRICITY PROBLEMS IN INDUCTION MOTORS			
MCA-1 (25 PAGE BOOKLET)		$ 39.00	
DC MOTOR & CONTROL PROBLEM DETECTION USING VIBRATION ANALYSIS			
DCM-1 (30 PAGE BOOKLET)		$ 49.00	
HOW TO DIAGNOSE PROBLEMS WITHIN ROTARY SCREW AIR COMPRESSORS HAVING MANY GEARS & BEARINGS USING VIBRATION AS WELL AS HIGH FREQUENCY DEMODULATED SPECTRAL ANALYSIS			
RSC-1 (60 PAGE BOOKLET)		$ 69.00	
TOTAL OF PAGE 1			

PLEASE COMPLETE PAGE 2 OF ORDER FORM

TECHNICAL ASSOCIATES OF CHARLOTTE, P.C.

SINCE 1961

347 N. Caswell Road Charlotte, NC 28204 Ph: (704) 333-9011 Fax: (704) 333-1728 www.technicalassociates.net

PUBLICATION ORDER FORM

	QTY.	PRICE	TOTAL
DESCRIPTION OF PEAKVUE & ILLUSTRATION WIDE ARRAY OF APPLICATIONS IN FAULT DETECTION & PROBLEM SEVERITY ASSESSMENT (NEW)			
PKV-1 (95 PAGE BOOKLET)	____	$ 89.00	____
HIGH-SPEED CENTRIFUGAL AIR COMPRESSOR DAIGNOSTICS			
CAC-1 (15 PAGE BOOKLET)	____	$ 29.00	____
ENTRY LEVEL VIBRATION ANALYSIS AND PREDICTIVE MAINTENANCE TECHNIQUES			
VSA-0 (200 PAGE SEMINAR BOOK)	____	$ 200.00	____
VIBRATION SIGNATURE ANALYSIS I			
VSA-1 (410 PAGE SEMINAR BOOK)	____	$ 250.00	____
VIBRATION SIGNATURE ANALYSIS II			
VSA-2 (490 PAGE SEMINAR BOOK)	____	$ 275.00	____
VIBRATION SIGNATURE ANALYSIS III			
VSA-3 (510 PAGE SEMINAR BOOK)	____	$ 300.00	____
ADVANCED VIBRATION DIAGNOSTIC AND REDUCTION TECHNIQUES			
ADV-1 (1150 PAGE SEMINAR BOOK)	____	$ 450.00	____
PREDICTIVE MAINTENANCE PROGRAM MANAGEMENT			
PMP-1 (250 PAGE SEMINAR BOOK)	____	$ 125.00	____
DYNAMIC FIELD BALANCING			
DFB-1 (100 PAGE SEMINAR BOOK)		$ 150.00	

NOTE: THE ABOVE PRICES INCLUDE UPS SHIPPING AND HANDLING WITHIN THE USA

FOR DELIVERY IN N.C., ADD 7% SALES TAX OR ADD SHIPPING CHARGES OUTSIDE USA ____

TOTAL FROM PAGE 1 ____

TOTAL FROM PAGE 2 ____

TOTAL ORDER (US DOLLARS) ____

BILLING INFORMATION

DATE ____

P.O. NUMBER: ____

CREDIT CARD NUMBER ____

☐ MASTERCARD ☐ VISA ☐ AMEX

EXP. DATE: ____

CARD HOLDER NAME: ____

ORDER PLACED BY: ____

PHONE NUMBER: ( ) ____

SHIP TO:

COMPANY NAME ____

ATTN: ____

ADDRESS ____

____

CITY STATE ZIP ____

BILL TO:

COMPANY NAME ____

ATTN: ____

ADDRESS ____

____

CITY STATE ZIP ____

Technical Associates offers vibration diagnostic investigation services and support for all phases of your vibration program from the initial startup throught the analysis of the collected data.

R-122001-01

CHAPTER 1

ANALYSIS I SEMINAR OVERVIEW

An effective Predictive Maintenance Program (PMP) is a total program of the following:

1. DETECTION
2. ANALYSIS
3. CORRECTION
4. VERIFICATION

This is a logical sequence of steps. The program first helps you detect the onset of a problem. It then provides means for analyzing the problem in order to determine its cause. It puts you in a position to correct the problem, effectively and efficiently, at a convenient time. And finally, it gives you a means to verify that any correction taken did in fact correct the problem and that no other problems were included.

Predictive Maintenance uses the process of tracking vibration levels on equipment components to determine the condition of the machinery. The guiding purpose of this seminar is to provide instruction on how vibration signature analysis can be used to continually evaluate machine condition within a Predictive Maintenance Program.

Predictive Maintenance Programs begin with Baseline (or initial) surveys of machines. Later, followup surveys are conducted at periodic intervals dependent on machine type, criticality, operating and maintenance cost, operating speed and design of components within such as bearing type, gearing type, etc. Following either Baseline or Followup Periodic Survey measurements, an analysis of collected data is made and written survey reports are compiled summarizing diagnostic results as well as providing overall recommendations such as those which follow:

1) No Problem Found.

2) Minor Problems Found - Trend Only at this Time During Future PMP Surveys.

3) Potentially Serious Problems Detected Which Might Tend to Deteriorate Within Weeks - Continue to Monitor Equipment at Shorter Intervals.

4) Potentially Serious Problems Detected, but Problem Source Not Yet Confirmed - Perform Vibration Diagnostics to Determine Problem Source and Severity.

5) Significant Problem(s) Detected Which Warrant Corrective Action - Replace Parts at Next Scheduled Shutdown.

6) Very Severe Problem(s) Detected Which Mandate Corrective Action - Shut Down and Replace Immediately.

© Copyright 2001 Technical Associates of Charlotte, P.C.

One of the overall purposes of this seminar will be to teach the student how to begin making these diagnostic calls using vibration signature analysis, how to evaluate problem severities and what thought process should be used to either recommended further diagnostic tests, or how to go ahead and recommend what corrective actions should be made in which chronological order.

Importantly, the analyst will then be instructed to verify all problems have been resolved (and no new ones have been introduced) after corrective actions have been taken. In other words, he should repeat his analysis to assess the machine condition after completion of such repairs. Documenting what actions were taken,along with the date they were completed, is an essential step to any successful program. However, taking the extra step to document the "Before" and "After" condition is what is critically needed to keep the program visible and effective in the minds of those associated, not only with Maintenance and Plant Management, but also to those in the Production Department.

Following the completion of this course, the user will have a good working knowledge of the proper application instrumentation and software necessary for setting up of and implementing an effective predictive maintenance program as well as receive instruction on basic concepts involved in troubleshooting vibration problems. The following are brief introductions for each of the chapters covered in this seminar text.

- CHAPTER 2 - "WHAT IS VIBRATION AND HOW CAN IT BE USED TO EVALUATE ` MACHINE CONDITION?": describes the basic theory of vibration which includes amplitude, frequency and phase. It also introduces the time waveform and how it is converted into a vibration spectrum. The relationships between acceleration, velocity, and displacement are illustrated, as well as their direct relation to machine wear, fatigue, and failure. The true meaning of Overall Vibration is also given here, along with how it is measured. This section also points out the great difference in whether this overall level is directly measured from a time waveform (Analog method), or if it is calculated from only the data within the spectrum itself between F_{MIN} and F_{MAX} (Digital method). It shows how much of a difference this can make simply by which method is employed. Also included are examples of trending vibration levels on equipment and determining vibration severity.

- CHAPTER 3 - "OVERVIEW OF THE STRENGTHS AND WEAKNESSES OF TYPICAL VIBRATION INSTRUMENTS": introduces the user to the types of vibration measurement equipment commonly found in a Predictive Maintenance Program as well as the vibration instruments and software required to analyze and diagnose vibration problems. A brief history of the development and enhancement of these tools is provided, along with updated information on some of the more state-of-the-art PMP systems available today.

- CHAPTER 4 - "OVERVIEW OF VARIOUS VIBRATION TRANSDUCERS AND THEIR OPTIMUM APPLICATIONS": describes the various transducers (vibration pickups) that measure the vibration oscillation and transmit this information back to the measuring instrument. Transducers are covered which can directly measure any of the three primary vibration parameters including acceleration, velocity, or displacement. Information on which particular transducer should be employed based on machine speed, machine components and overall frequency response is also covered.

- CHAPTER 5 - "ROLE OF SPIKE ENERGY, HIGH FREQUENCY DETECTION (HFD) AND SHOCK PULSE (SPM), ALONG WITH PROPER SPECIFICATION OF THEIR ALARM LEVELS AT VARIOUS SPEEDS": describes the theory behind each of these parameters and illustrates the types of machinery problems that are and are not responsive to these high frequency signals. Acceptable spike energy, HFD and shock pulse levels are provided from severity charts which have been developed for these parameters.

© Copyright 2001 Technical Associates of Charlotte, P.C.

- CHAPTER 6 - "INTRODUCTION TO VIBRATION SIGNATURE ANALYSIS AND HOW IT IS USED TO DIAGNOSE MACHINE OPERATING CONDITION": introduces the user to Technical Associates' renown 5-Page "Illustrated Vibration Diagnostics Chart" which shows the user a typical spectrum for each of 48 problem conditions (unbalance, misalignment, rolling element bearing problems, etc.), demonstrates phase response for those problems which have an effect on phase, and provides supporting remarks summarizing the more important symptoms each problem will normally display. Following presentation of the diagnostic chart will be discussions separately covering many of the problem conditions listed in the chart. However, since this seminar is intended to be an introductory course, only the more common problems will be covered (all remaining chart items are covered in the next seminar module). Included in many of the theory discussions are vibration spectra taken from machines having the particular problem being covered. Finally, real-world case histories representing each of the problems covered in Chapter 6 are included in the last chapter of the seminar text.

- CHAPTER 7 - "PROVEN METHOD FOR SPECIFYING SPECTRAL BAND ALARM LEVELS AND FREQUENCIES USING TODAY'S PREDICTIVE MAINTENANCE SOFTWARE SYSTEMS": illustrates how to properly specify not only overall vibration alarms for each point on each machine in a program, but also how and why various portions of each spectrum should be set apart, allowing much more for some of these so-called "spectral alarm bands" than for others. For example, if one band is set around operating speed (1X RPM), whereas another band is specified around bearing defect frequencies for the machine, much more vibration is most always allowed for 1X RPM as compared to that for bearing defect frequencies (which are covered in Chapter 6). Note in this chapter that different machine types will have a different spectral band setup (for example, a pump having a 6-vaned impeller and rolling element bearings versus a motor outfitted with sleeve bearings). These spectral bands are available in several Predictive Maintenance Software systems. The vibration amplitudes within these bands are summed to produce a so-called "power" level in each band which is compared with an allowable level for that band specified by the user. Information is given on how this so-called "RSS Power Level" is calculated for each band and why this is important when attempting to specify meaningful alarm levels for each spectral band. Details of the starting and ending frequencies for each band are discussed, along with the band alarm as per the type of equipment being monitored. Also provided is an important parameter - where the maximum frequency (F_{MAX}) should be set for each type of machine, bearing type and operating speed.

- CHAPTER 8 - "COMMON PITFALLS IN EVERYDAY VIBRATION MEASUREMENTS AND THEIR EFFECT ON PROPERLY DIAGNOSING MACHINE PROBLEMS": informs the user on the proper choice and use of vibration transducers based on transducer mounting, frequency range, and output parameter (acceleration, velocity, or displacement) desired. The pros and cons of the vibration measurement location (bearing housing, shaft, etc...) are also evaluated. It also clearly demonstrates the detrimental effect on accuracy of diagnostic calls if the wrong transducer (or transducer mount) is used; if the wrong frequency span is measured; if measurements are made from the wrong location on a machine; etc.

- CHAPTER 9 - "SETUP AND IMPLEMENTATION OF EFFECTIVE PREDICTIVE MAINTENANCE AND CONDITION MONITORING PROGRAMS": pulls together the information on all the PMP hardware and software tools, signature analysis and overall/spectral band alarm settings covered in the previous chapters and shows the user how to use this knowledge to put together a complete and cost effective Predictive Maintenance Program (PMP). Implementation costs vs. payback savings are presented. A flow chart describing the steps necessary to create an effective Predictive Maintenance Program is presented, as well as the necessary data sheets that are required for determining proper alarm settings, frequency ranges, measurement parameters (i.e., acceleration, velocity, displacement, etc.), as well as how to determine optimum data collection routes. Examples of machine condition reports are provided which include frequency spectra from damaged machines; overall condition for each machine evaluated;

© Copyright 2001 Technical Associates of Charlotte, P.C.

severity rank-ordered reports for problem machines; alarm exception reports; and machine measurement reports which show the overall level measured and recorded for each point on each machine evaluated.

- CHAPTER 10 - "ACTUAL CASE HISTORIES OF VIBRATION DIAGNOSTIC ON VARIOUS MACHINE TYPES": offers an array of real-world problems which have been detected by Predictive Maintenance using the tools taught in this seminar. They show how problems were determined by vibration diagnostics, and subsequently corrected without catastrophic failure. Impressive "before" and "after" frequency spectra are displayed to show the effect of completing the recommended corrective actions on the machine, and thereby prolonging the life of the equipment by reducing vibration amplitudes.

SEMINAR GOALS

At the conclusion of this seminar, it is hoped the user of this manual will be able to effectively communicate on a technical level with others in the field of vibration analysis. In addition, he should also gain a fundamental understanding of how to effectively start up and fully implement an effective Predictive Maintenance Program. Furthermore, an introduction into the use of vibration analysis for purposes of troubleshooting equipment problems will have been presented by this manual. Your questions and comments pertaining to this course's educational content are welcome and will provide the basis for enhancements to better train future seminar students.

© Copyright 2001 Technical Associates of Charlotte, P.C.

CHAPTER 2

WHAT IS VIBRATION AND HOW CAN IT BE USED TO EVALUATE MACHINERY CONDITION?

2.0 INTRODUCTION

Vibration is defined by Webster's New World Dictionary as "to swing back and forth; to oscillate". For those more closely associated with the vibration industry, vibration is a pulsating motion of a machine or a machine part from its original place of rest and can be represented by this formula:

$$\text{VIBRATION AMPLITUDE RESPONSE} \propto \frac{\text{DYNAMIC FORCE}}{\text{DYNAMIC RESISTANCE}} \qquad \text{Equation 1}$$

A good way to illustrate vibration of a machine is shown in Figure 1 below. Equation 1 shows that vibration amplitude varies with the quotient of Dynamic Force divided by Dynamic Resistance (that is, a machine with acceptable vibration may have a marked higher level if given insufficient support frame and foundation).

Vibration is the response of a system to some internal or external stimulus or force applied to the system. Vibration has three important parameters which can be measured - Amplitude (how much); Frequency (how many times per minute or per second); and Phase (which describes how it is vibrating). Each of these important parameters will be discussed in sections of Chapter 2 which follow.

Note that a vibration transducer is mounted on (or as near as possible to) the bearing housing. This transducer will sense the vibration and pass the signal through a connecting cable to an analyzer. Figure 1 shows that the machine's bearing housing can be modeled by a mass suspended by a coil spring. Until a force (stimulus) is applied to this mass, it will remain suspended in a neutral or unstimulated position. When a force is applied to the mass (i.e., in an upward direction in this case) as shown in Figure 2, the mass moves upward and the spring compresses (stimulated) by the force.

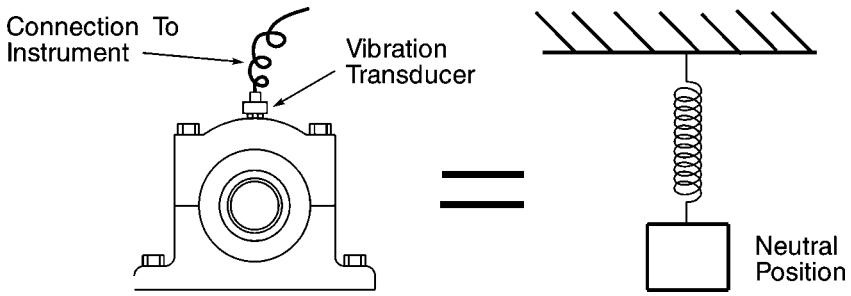


FIGURE 1
MASS IN NEUTRAL POSITION WITH NO APPLIED FORCE

© Copyright 2001 Technical Associates Of Charlotte, P.C.

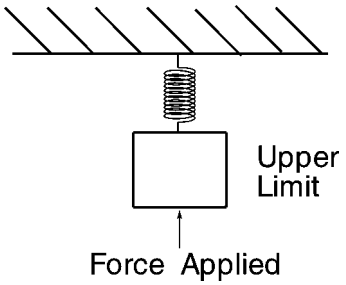


FIGURE 2
MASS BEING STIMULATED BY AN APPLIED FORCE

Once an upper limit of motion is reached the force is removed and the mass begins to drop. The mass will drop through the neutral position and continue to travel to its lower limit as shown in Figure 3.

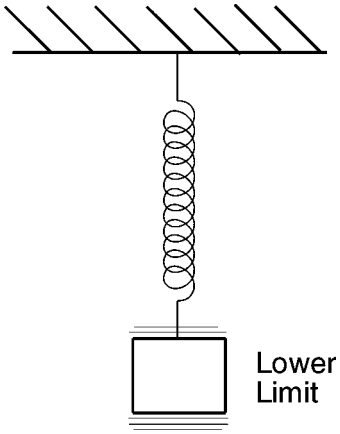


FIGURE 3
MASS RESPONDING TO THE RELEASE OF THE APPLIED FORCE

Once the lower limit is reached, the mass will stop its downwards motion and reverse direction again passing through the neutral position to the upper limit; then stop and return to the lower limit repetitively as long as an external force is applied as shown in Figure 4.

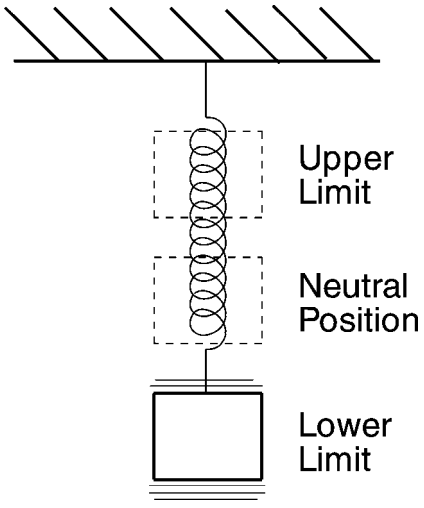


FIGURE 4
CONTINUED RESPONSE TO APPLIED FORCE

© Copyright 2001 Technical Associates Of Charlotte, P.C.

If a pen were attached to the mass as it repetitively responds to the applied force and a strip chart recorder placed nearby, the vibration response could be documented as shown in Figure 5.

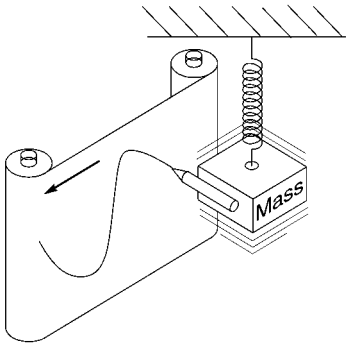


FIGURE 5
PEN ADDED TO MASS TO TRACE ITS OSCILLATING MOTION ON A CONSTANT SPEED STRIP CHART RECORDER

2.01 WHAT IS VIBRATION FREQUENCY AND HOW DOES IT RELATE TO A TIME WAVEFORM?

When examined, the trace drawn on the strip chart recorder of Figure 5 would show a uniform sine wave having an amplitude (peak-to-peak displacement in this case). Frequency can be calculated from it by measuring the time period (T) of one cycle and inverting to determine the frequency. See Figure 6. Frequency is expressed in units of either Cycles per Minute (CPM) or in Cycles per Second (CPS), which is now called Hertz (where 1 Hertz = 60 CPM). The commonly used abbreviation for Hertz is “Hz”.

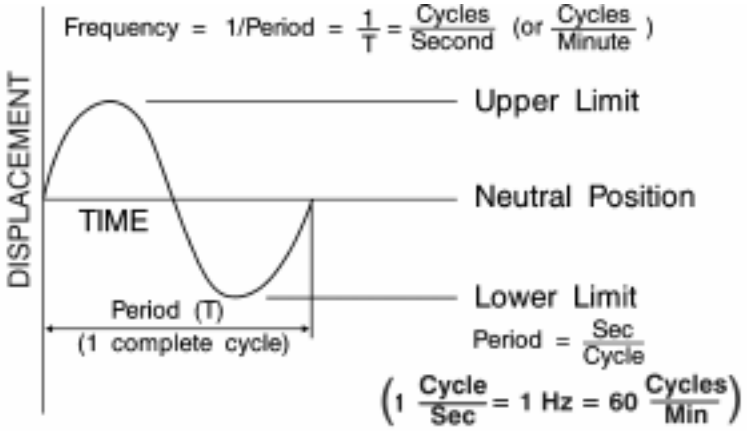


FIGURE 6
DISPLACEMENT AND FREQUENCY FROM THE TIME WAVEFORM

2.02 WHAT IS VIBRATION DISPLACEMENT?

As its name implies, displacement is a measure of the total travel of the mass - that is, it shows how far the mass travels back and forth when it vibrates. Displacement of the mass can be expressed either in units of mils (where 1 mil = .001 inch) or in microns (where 1 micron = .001 millimeter). Further extrapolation of the same displacement sine waveform would yield velocity and acceleration values as seen in Figures 7 and 8.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

2.03 WHAT IS VIBRATION VELOCITY?

Vibration velocity is a measure of the speed at which the mass is moving or vibrating during its oscillations. The speed of the mass is zero at the upper and lower limits since it must come to a stop at these points before it reverses direction and begins to move in the opposite direction. Velocity reaches its maximum (or peak) at the neutral position where the mass has fully accelerated and now begins to decelerate as shown in Figure 7. Velocity is expressed in units of inches per second (in/sec) or millimeters per second (mm/sec).

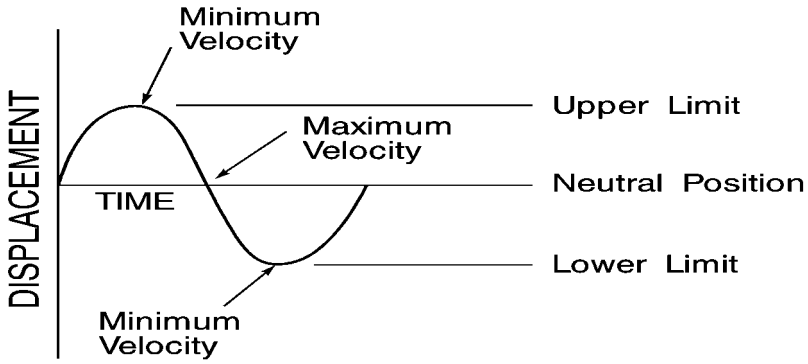


FIGURE 7
VELOCITY FROM THE DISPLACEMENT WAVEFORM

2.04 WHAT IS VIBRATION ACCELERATION?

Acceleration is defined as the rate of change of the velocity and is measured in g's of acceleration relative to the acceleration of gravity. At sea level, 1.0 g equals 32.2 ft/sec^2 which equals 386.087 in/sec/sec or 9806.65 mm/sec/sec - the accepted values for the acceleration of gravity in the English and Metric systems (where in/sec/sec is normally expressed as in/sec^2). Acceleration as noted in Figure 8 is greatest where velocity is at a minimum. This is where the mass has decelerated to a stop and is about to begin accelerating again (that is, moving faster).

Acceleration is probably the most difficult measure of vibration amplitude to grasp. To bring it into clearer focus, let us return to the mid-1950's and to a group of seven men having the "right stuff" and follow them on their journey towards the center of the earth from within a specially designed compartment in a jet plane. When the plane began its descent at approximately 400 miles per hour and started increasing speed at the rate of about 26 miles per hour each second (or 386 in/sec/sec), these men began to experience that temporarily wonderful feeling of weightlessness as the plane overcame the force of gravity. That is, when the plane accelerated at this rate (400 miles per hour in the beginning; then 426 MPH one second later; 452 MPH the next second and so on), a "force" of 1.0g was applied to each man whether he weighed 150 lbs, or 250 lbs, and each man began to float around within the cabin. This was all well and good until... it was time for the plane to pull out of its downward plunge (of course, the alternative was much less fun). When the plane pulled out of the dive, each man was then subjected to acceleration levels of 6g or more (that is, the skeletal structure of a 200 lb man suddenly had to withstand a force of roughly 1200 lbs!).

In the same way, when a machine housing vibrates, it experiences the force of acceleration since it continually changes speed as it moves back and forth. The greater the rate of change of this speed (or velocity), the higher will be the forces on this machine due to the higher acceleration. Therefore, the greater this amount of acceleration, the higher will be the forces (and thus, stresses) applied to the vibrating machine member.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

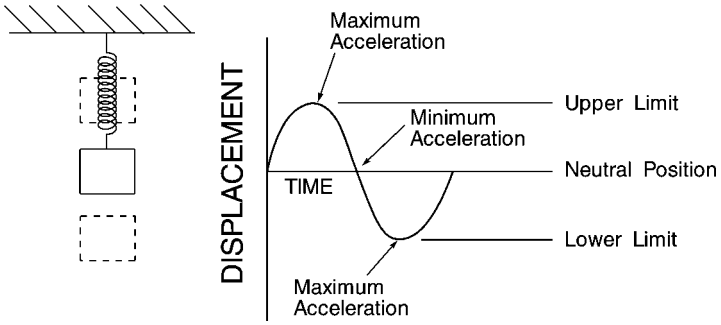


FIGURE 8
ACCELERATION FROM THE DISPLACEMENT WAVEFORM

2.05 WHAT IS VIBRATION PHASE?

Vibration phase is the final descriptive characteristic of vibration. Phase is the relative shift of a vibrating part to a fixed reference point on another vibrating part. That is, phase is a measure of the vibration motion at one location relative to the vibration motion at another location. Or, in other words, it is the “timing” of a vibration in relation to a stationary or moving part on the machine. This is similar to the “timing” reference used to tune a reciprocating engine. Phase is a powerful tool useful in the analysis of machine faults which will be discussed later. Since phase is a measure of relative motion, examples can be shown with two mass weights and springs attached to the same reference point. Figure 9 shows the two systems in-phase with each other or vibrating at the same rate with 0° phase difference and the resulting time waveform.

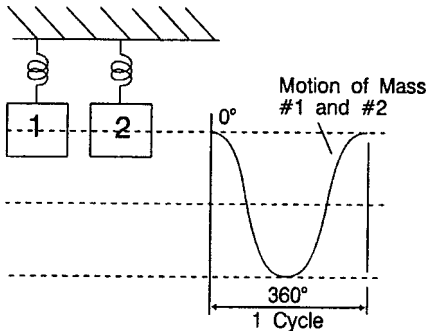


FIGURE 9
TWO MASSES WITH ZERO PHASE DIFFERENCE

Figure 10 shows two masses vibrating with 90° phase difference. That is, Mass #2 is one-fourth of a cycle (or 90°) ahead of Mass #1. In general vibration literature, this is what is meant when someone states “Mass #1 has a 90° phase lag relative to the motion of Mass #2".

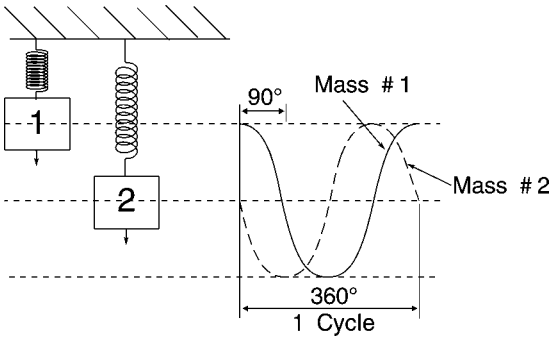


FIGURE 10
TWO MASSES WITH 90° PHASE DIFFERENCE

© Copyright 2001 Technical Associates Of Charlotte, P.C.

In the same way, Figure 11 shows the same two masses vibrating with a 180° phase difference. That is, at any instant of time, Mass #1 moves downwards at the same time as Mass #2 moves upwards, and vice versa.

Vibration phase is measured in angular degrees by using a strobe light or electronic photocell.

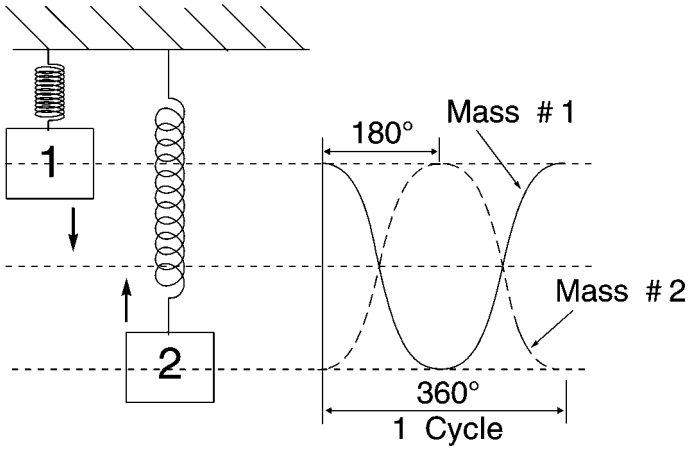


FIGURE 11
TWO MASSES WITH 180° PHASE DIFFERENCE

Figure 12 shows how phase relates to machine vibration. The left sketch shows a 0° phase difference between bearing Positions 1 and 2 (in-phase motion); while the right sketch pictures a 180° out-of-phase difference between these positions (out-of-phase motion).

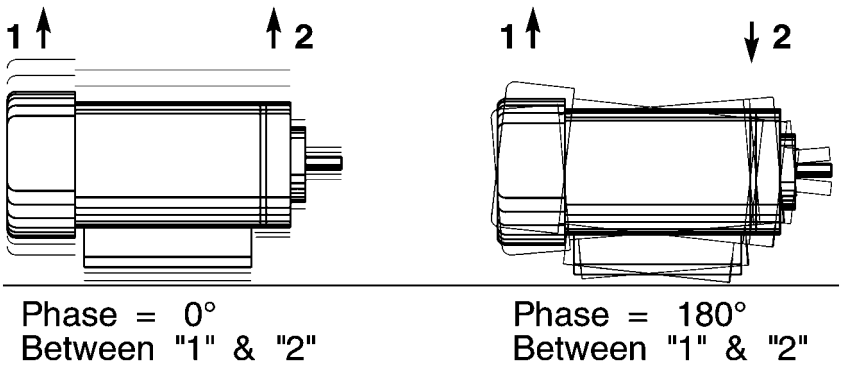


FIGURE 12
PHASE RELATIONSHIP AS USED WITH MACHINERY VIBRATION

© Copyright 2001 Technical Associates Of Charlotte, P.C.

2.1 WHAT IS A VIBRATION SPECTRUM (ALSO CALLED AN “FFT” OR “SIGNATURE”)?

A vibration spectrum plots or graphs amplitude (mils, in/sec or g’s) versus frequency (CPM or Hz) as shown in Figure 16c. Getting back to the so-called vibration measurement system shown in Figure 5, note that this direct recording method has many limitations. To overcome these problems, transducers are used to convert vibration into an electrical signal. This electrical signal is then translated through electronics into a vibration display within an analyzer to which the transducer is connected.

There are many shortcomings to using the aforementioned direct recording method to measure vibration data. A much more effective method is to use electronics which is used to convert the vibration sensed by the transducer itself into an electronic signal. One such setup is shown in Figure 13 in which the transducer signal travels through an amplifier to a servo motor which drives a pen to graph the vibration motion on a chart recorder.

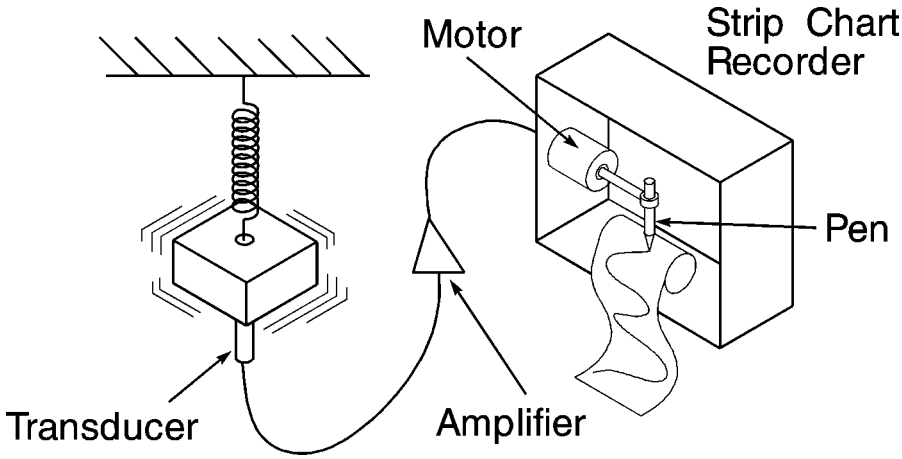
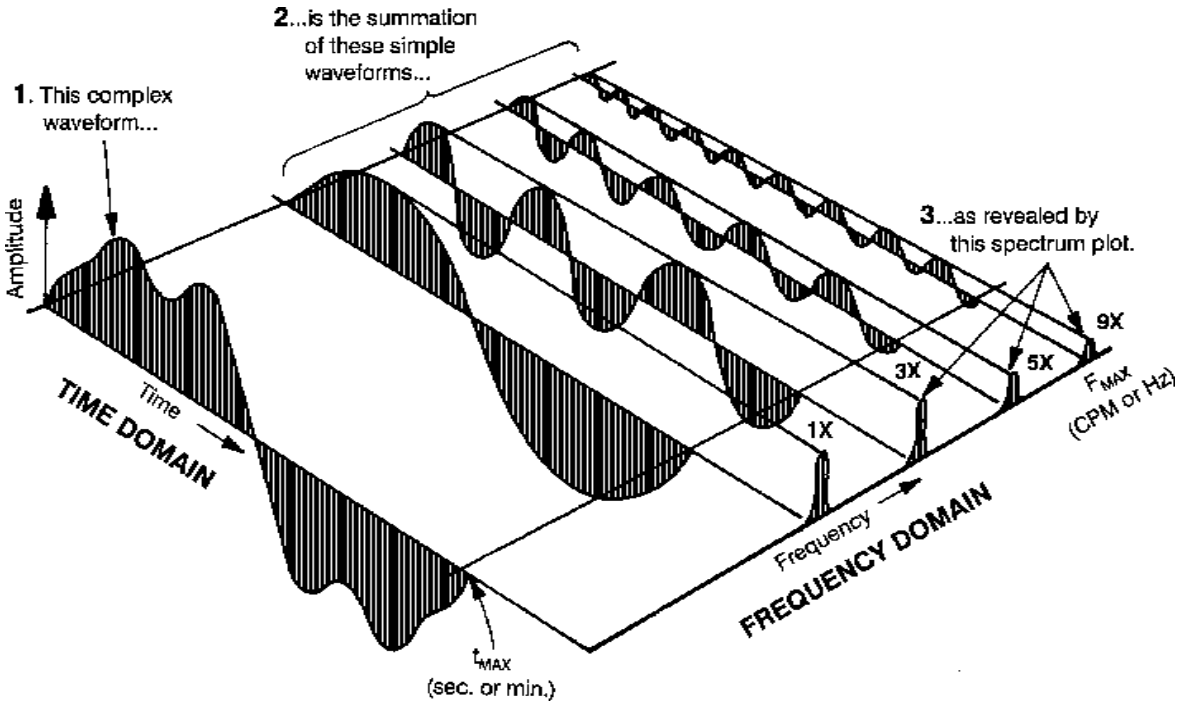


FIGURE 13
INDIRECT MEANS OF RECORDING VIBRATION

The amplitude of the recorded waveform can be adjusted to be the same as that of the actual vibrating piece by adjusting the gain of the amplifier. Figure 13 is a very simplified illustration of vibration recording. However, most vibrations are complicated combinations of various waveforms that require more sophisticated indirect recording devices. Figure 14 shows how waveforms combine to make more complicated waveforms. Note how the total waveform is actually made up of a series of smaller waveforms, each of which correspond to an individual frequency (1X RPM, 2X RPM, 3X RPM, etc.). Each of these individual waveforms then algebraically add to one another to generate the total waveform which can be displayed by one of today’s oscilloscopes or analyzers.

An oscilloscope is useful in viewing these combined waveforms. It functions by passing the signal from a transducer into two electronic plates that are able to displace an electronic beam into the shape of the waveform. Figure 15 illustrates this process.

© Copyright 2001 Technical Associates Of Charlotte, P.C.



t_{MAX} (sec) $= \frac{(60)(\#\ FFT\ Lines)}{F_{MAX}} = \frac{(60)(Sample\ Size)}{(2.56)(F_{MAX})}$	F_{MAX} (CPM) $= \frac{(60)(\#\ FFT\ Lines)}{t_{MAX}} = \frac{(60)(Sample\ Size)}{(2.56)(t_{MAX})}$

Where: t_{MAX} = Total Sampling Period setting how often amplitude is measured and stored (sec).
Sample Size = Number of Analog to Digital Conversions to be used to Construct the Time Waveform (Samples- most often 1024 Samples to provide a 400 line FFT).
F_{MAX} = Maximum Spectral Frequency or Frequency Span (CPM).

FIGURE 14
COMPARISON OF TIME & FREQUENCY DOMAINS (Ref. 14)

Until now in this paper, all the illustrated measurements have been shown in the time domain where the X-axis is time (sec or min.) while the Y-axis is a vibration amplitude measure (displacement, velocity or acceleration). Displaying and using the time domain is a very precise method in which to display the actual (total) machine motion and to analyze the various vibration parameters. However, analyzing the time waveform itself can be very cumbersome and labor intensive when frequency needs to be determined. Here, the time duration from one peak of interest to the next similar peak must be determined to calculate the vibration period (sec/cycle) as seen in Figures 6 and 16. This cycle time or period (T) must then be inverted to obtain frequency (F) and then converted to the proper frequency units (CPM, CPS, Hertz).

To simplify this process, vibration instruments are able to develop what is known as a Fast Fourier Transform (FFT). An FFT is the computer (microprocessor) transformation from time domain data (amplitude versus time) into frequency domain data (amplitude versus frequency). This FFT calculation technique was developed by Baron Jean Baptiste Fourier over 100 years ago. Fourier stated that any real-world sine waveform can combine to make another more complicated waveform as was seen in Figure 14; and vice versa, any complicated real-world waveform can be separated into its simple sine waveform components. This is illustrated in Figure 16 where: (a) a total time waveform is captured in the time domain; where in (b) this time domain waveform is separated into its separate sine waveforms and displayed in three dimensional coordinates of amplitude, time and frequency. As the sine waves are separated from the combined waveform, the frequency of each sine wave is determined and the sine waves are

© Copyright 2001 Technical Associates Of Charlotte, P.C.

placed in the respective positions along the frequency axis. In Figure 16C, a view looking along each individual waveform is shown allowing the analyst to see the frequency domain of amplitude versus frequency.

The frequency domain view (Figure 16C) of the time waveform (Figure 16A) visually shows each simple sine wave as a vertical line that has amplitude (as determined by its height) and frequency (as determined by its position along the frequency axis). This frequency domain representation of a time waveform is called a spectrum (with spectra being its plural form). A spectrum is sometimes referred to as a "signature" or an "FFT". Spectra (frequency domain displays) are very useful tools for the vibration analyst who would otherwise have the laborious task of distinguishing and separating time waveforms into discrete frequency components for analysis.

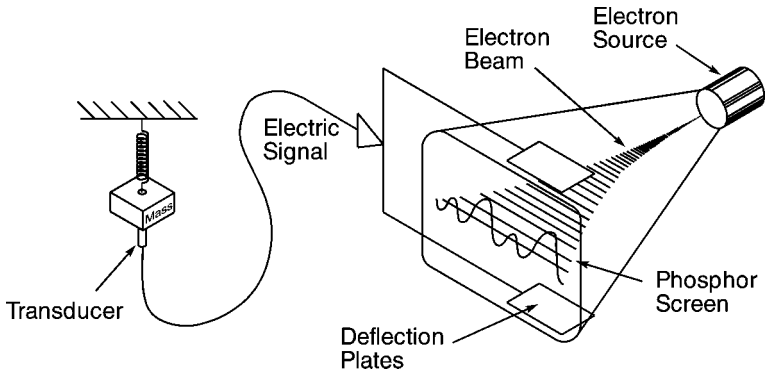
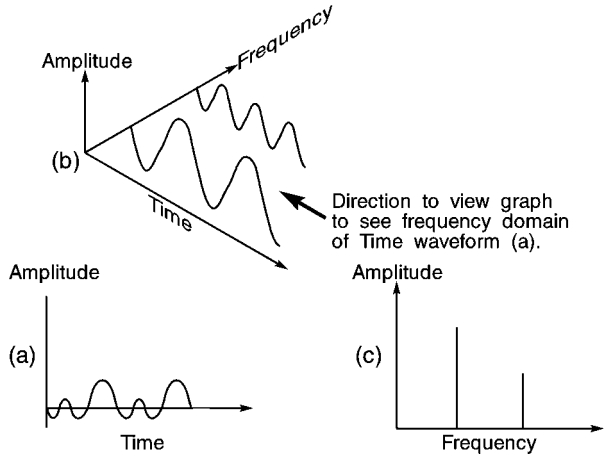


FIGURE 15
VIBRATION SIGNAL DISPLAYED ON AN OSCILLOSCOPE



a) Captured time domain waveform

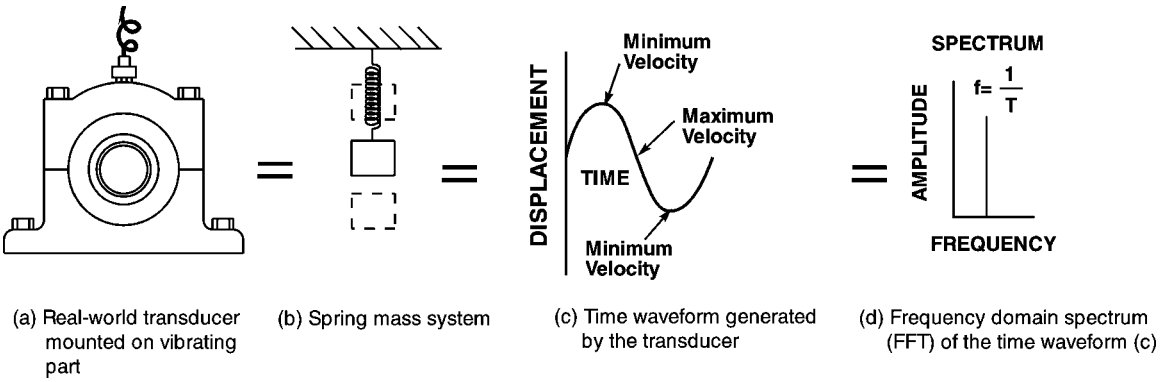
b) Three-dimensional view of the captured time domain waveform after separation into its separate simple sine waves and determination of the frequency of each sine wave. The three axes are amplitude, time and frequency.

c) Frequency domain view developed from the captured time waveform(s).

FIGURE 16
FREQUENCY DOMAIN DEVELOPED FROM THE TIME DOMAIN WAVEFORMS

© Copyright 2001 Technical Associates Of Charlotte, P.C.

Figure 17 illustrates the full transformation of vibration as equated to a real-world transducer mounted on a bearing housing (a) as represented by the spring mass system (b) capturing a time waveform of the vibration (c) which is then processed into an FFT spectrum in the frequency domain (d).



(a) Real-world transducer mounted on vibrating part

(b) Spring mass system

(c) Time waveform generated by the transducer

(d) Frequency domain spectrum (FFT) of the time waveform (c)

FIGURE 17
STEPS IN THE CONVERSION OF A VIBRATION INTO AN FFT SPECTRUM

© Copyright 2001 Technical Associates Of Charlotte, P.C.

2.2 DIFFERENCE BETWEEN RMS, PEAK AND PEAK-TO- PEAK VIBRATION AMPLITUDE?

Since electronics is already employed to create the various vibration displays in the time and frequency domains, additional electronics and software are used to convert the displayed amplitude into measurement units of displacement, velocity and acceleration. The electronics also performs all the necessary conversions for peak to peak, peak, RMS (root-mean-square) or average amplitude. Table I gives the formulas for the various measurement unit conversions. It is interesting to note here that Europeans normally use RMS velocity amplitudes while Americans have adopted peak velocity values even though the instruments themselves collect RMS data and then multiply them by the conversion factor (1.414) to obtain so-called peak velocity. This is most likely due to the fact that most all the severity charts for various equipment types have been developed using this so-called peak velocity in America.

Figure 18 compares the various common English and Metric units of vibration measurement.

TABLE I
CONVERSION FORMULAS FOR VARIOUS AMPLITUDE UNITS (Ref. 1)
(See Figure 22 also which graphically shows these and gives further formulas)

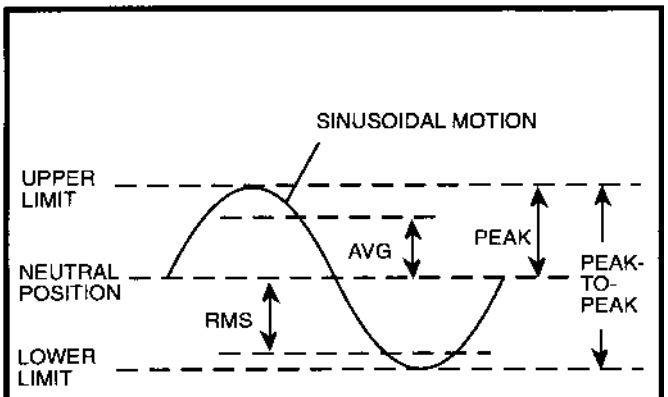
ENGLISH	METRIC
$D = 19{,}100\ (V)/F$	$D_M = 19{,}100\ (V_M)/F$
$D = 70{,}471{,}000\ (A)/F^2$	$D_M = 1.79 \times 10^9\ (A)/F^2$
$V = DF/19{,}100$	$V_M = D_M F/19{,}100$
$V = 3690\ (A)/F$	$V_M = 93{,}580\ (A)/F$
$A = DF^2/70{,}471{,}000$	$A = D_M F^2/1.79 \times 10^9$
$A = VF/3690$	$A = V_M F/93{,}580$
A= Acceleration (g PK) V= Velocity (in/sec PK) D= Displacement (mils PK-PK) F= Frequency (CPM)	A= Acceleration (g PK) V= Velocity (mm/sec PK) D= Displacement (microns PK-PK) F= Frequency (CPM)

VIBRATION CHARACTERISTICS	COMMON UNITS OF MEASUREMENT	
	ENGLISH	METRIC
FREQUENCY	CPM OR Hz	CPM OR Hz
DISPLACEMENT	MILS PEAK-TO-PEAK	MIICRONS, PEAK
VELOCITY	IN/SEC RMS IN/SEC PEAK	MM/SEC RMS MM/SEC PEAK
ACCELERATION	G PEAK	G PEAK
PHASE	DEGREES	DEGREES

FIGURE 18
COMPARISON OF ENGLISH AND METRIC VIBRATION UNITS

© Copyright 2001 Technical Associates Of Charlotte, P.C.

Figure 19 shows how one can convert from one unit of amplitude (i.e., RMS, Average, Peak and Peak-to-Peak) to another (for a pure sinusoidal sinewave which can be caused by unbalance for example). That is, if one found he had an amplitude unit which he needed to convert for comparison to another unit of amplitude, he would find the multiplier in Figure 19. For example, if one measured an amplitude of 1.0 in/sec RMS, this would equal 1.414 in/sec peak; whereas 1.0 mil RMS would equal 2.828 mils Pk-Pk.



MULTIPLY NUMBER OF → / BY / TO OBTAIN ↓	PEAK-TO-PEAK	PEAK	RMS	AVERAGE
PEAK-TO-PEAK	1.000	2.000	2.828	3.142
PEAK	.500	1.000	1.414	1.571
RMS	.354	.707	1.000	1.5
AVERAGE	.318	.636	.900	1.000

FIGURE 19
COMPARISON OF PEAK, PEAK-TO-PEAK, RMS, AND AVERAGE

Note from Figure 19 that for a sinusoidal waveform like that shown (which is likely caused by almost pure unbalance):

A. Peak-to-Peak Vibration = 2.000 X Peak Vibration
B. Peak-to-Peak Vibration = 2.828 X RMS Vibration
C. Peak Vibration = 0.500 X Peak-to-Peak Vibration
D. Peak Vibration = 1.414 X RMS Vibration
E. RMS Vibration = 0.354 X Peak-to-Peak Vibration
F. RMS Vibration = 0.707 X Peak Vibration

© Copyright 2001 Technical Associates Of Charlotte, P.C.

2.3 WHEN TO USE DISPLACEMENT, VELOCITY OR ACCELERATION?

When attempting to analyze a machine's vibration, it is essential that as much information as possible is available (bearing types and model numbers, accurate measures of the speed of each shaft, gear tooth counts, quantity of vanes or blades, etc.). Not knowing this information can seriously affect the accuracy of the diagnosis. Each one of these elements is very important in diagnosing vibration. Forces both inside and outside the machine result in some vibration at the machine. Sometimes the vibration is acceptable; at other times, the vibration is not acceptable and will eventually have an adverse affect on the machine health. Evaluating the vibration amplitude, frequency and phase can identify the condition the machine is in, both at the time of original baseline signatures as well as in followup Reliability Surveys.

Amplitude is one of the most frequently used parameters in vibration analysis. Vibration amplitude is proportional to the severity of potential machine problems and is one of the prime indicators of machine condition. Amplitude units of either displacement, velocity or acceleration can be used. Each of these units of measurement can be used. However, generally speaking, velocity is preferred.

Peak to peak displacement measurements are often the favored unit of measurement in the English system since displacement measurements relate directly to dial indicator measurements. Displacement is usually thought to be most useful in frequency ranges less than approximately 600 CPM (10 Hz). Frequency must be used along with displacement to evaluate vibration severity. That is, simply stating that the vibration at 1X RPM is 2 mils (25.4 microns Pk) is not enough information to relate if this 2 mil level is good or bad. For instance, 2 mils Pk-Pk of vibration at 3600 CPM is much more destructive than is the same 2 mils at 300 CPM (see Figure 20). Thus, displacement alone is unable to evaluate vibration severity throughout a full spectrum frequency range. This will be covered again below and recommended vibration severity charts for various machine types will be given.

Acceleration has similar disadvantages to displacement except it favors the higher frequency ranges. Acceleration is also frequency dependent in terms of severity or damage criteria. For example, 2g's at 18,000 CPM is much more severe than the same 2g's at a frequency of 180,000 CPM (3000 Hz) (see Figure 21). Acceleration is typically recommended for use when sources within a machine generate frequencies over approximately 300,000 CPM (5000 Hz). These sources may include gear mesh frequencies (# teeth X RPM), rotor bar passing frequencies (# rotor bars X RPM), blade passing frequencies (# blades X RPM), etc. Don't forget that in many cases, these sources generate harmonics (or multiples) of these frequencies.

Velocity on the other hand is not nearly so frequency dependent in the range from about 600 to 120,000 CPM (10 to 2000 Hz) and is generally the unit of choice when vibration sources within a machine generate frequencies ranging from approximately 300 to 300,000 CPM (5 to 5000 Hz). Velocity amplitudes relate directly to machine condition, no matter the frequency in the range from approximately 600 to about 100,000 CPM (10 to 1670 Hz). That is, it is generally felt a machine experiencing .30 in/sec at for example 1800 RPM is subjected to the same severity as another machine subjected to .30 in/sec at about 10,000 RPM operating speed. This is shown by the "Contours of Equal Severity" in Figure 22.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

2.4 HOW MUCH IS TOO MUCH VIBRATION?

Figure 20 is a typical displacement/velocity severity chart developed years ago for “general rotating machines” with vibration severity noted as being “GOOD”, “FAIR”, “ROUGH”, etc. This chart clearly illustrates the frequency dependence of displacement. For instance, a peak-to-peak displacement of 2 mils (25.4 microns, PK) ranges from “SMOOTH” to “VERY ROUGH” in severity. For this sample, frequency must also be identified to accurately determine severity (i.e., 2 mils of vibration @ 400 CPM is “GOOD” while 2 mils at 3600 CPM is rated “ROUGH”). Velocity on the other hand needs only its amplitude to determine severity on this chart (i.e., .157 to .314 in/sec of vibration is “SLIGHTLY ROUGH”). Figure 21 is a similar vibration severity chart for acceleration. Acceleration severity is again noted as being frequency dependent and velocity as independent of frequency. For example 2g’s at 18,000 CPM is rated “ROUGH”, while 2g’s at 180,000 CPM is rated just within the “GOOD” range.

Figure 22 shows the consistency which velocity has over a wide flat frequency range as compared to displacement and acceleration which tend to favor the low and high ends of the frequency scale respectively. Note that Figure 22 graphs “CONTOURS OF EQUAL SEVERITY” for each of the 3 amplitude measurements. Example A shows how to use equations given in Figure 22 to convert velocity to displacement, velocity to acceleration, displacement to acceleration, etc.

Figure 23 shows 3 spectra in (a) displacement, (b) velocity and (c) acceleration of the same waveform looking for a possible bearing defect problem.

Note the presence of the peak labeled 300 CPM in each of the 3 spectra. This is the frequency at operating speed (commonly called 1X RPM). However, note that as one moves from displacement to velocity into the acceleration spectrum, 1X RPM appears to be less and less a contributor. That is, looking at Figure 23A, note that 1X RPM clearly dominates the displacement spectrum. However, it is only slightly higher than 2 or 3 other peaks in the velocity spectrum of Figure 23B. Then, the 1X RPM peak almost vanishes in the acceleration spectrum of Figure 23C where its amplitude was just enough to reach the user-defined threshold so that its amplitude and frequency could be printed out on the plot.

Possibly of greater importance when detecting problems and evaluating machine condition is what happens to what are known as the bearing defect frequencies in this same figure (as the name implies, bearing defect frequencies are generated when wear begins to occur on either the bearing races, rolling elements or cage). Whether or not the analyst will see these important bearing frequencies in his spectra may depend on his choice of amplitude measure as demonstrated by Figure 23. With respect to the bearing frequencies at 4860 CPM and 9720 CPM, note that these frequencies are clearly visible on both the velocity and acceleration spectra (Figures 23B and 23C). What are known as sideband frequencies spaced at equal distances to the left and right of 4860 CPM, sideband frequencies can indicate a more serious bearing wear problem so it is important that the spectrum show their presence if they are there. Looking at Figure 23, note that while the 4860 CPM bearing frequency was clearly present in the velocity spectrum, it was just high enough in the displacement spectrum to reach the user-defined threshold. Of greater concern was the fact that the displacement spectrum missed the sideband frequencies surrounding 4860 CPM almost altogether, while it in fact did entirely miss the second bearing frequency at 9720 CPM. Of course, the reason for this is that displacement tends to “amplify” or emphasize low frequencies whereas it suppresses higher frequency peaks (as illustrated in Figure 22). On the other hand while acceleration emphasizes higher frequencies, it tends to suppress low frequencies (see Figure 22).

© Copyright 2001 Technical Associates Of Charlotte, P.C.

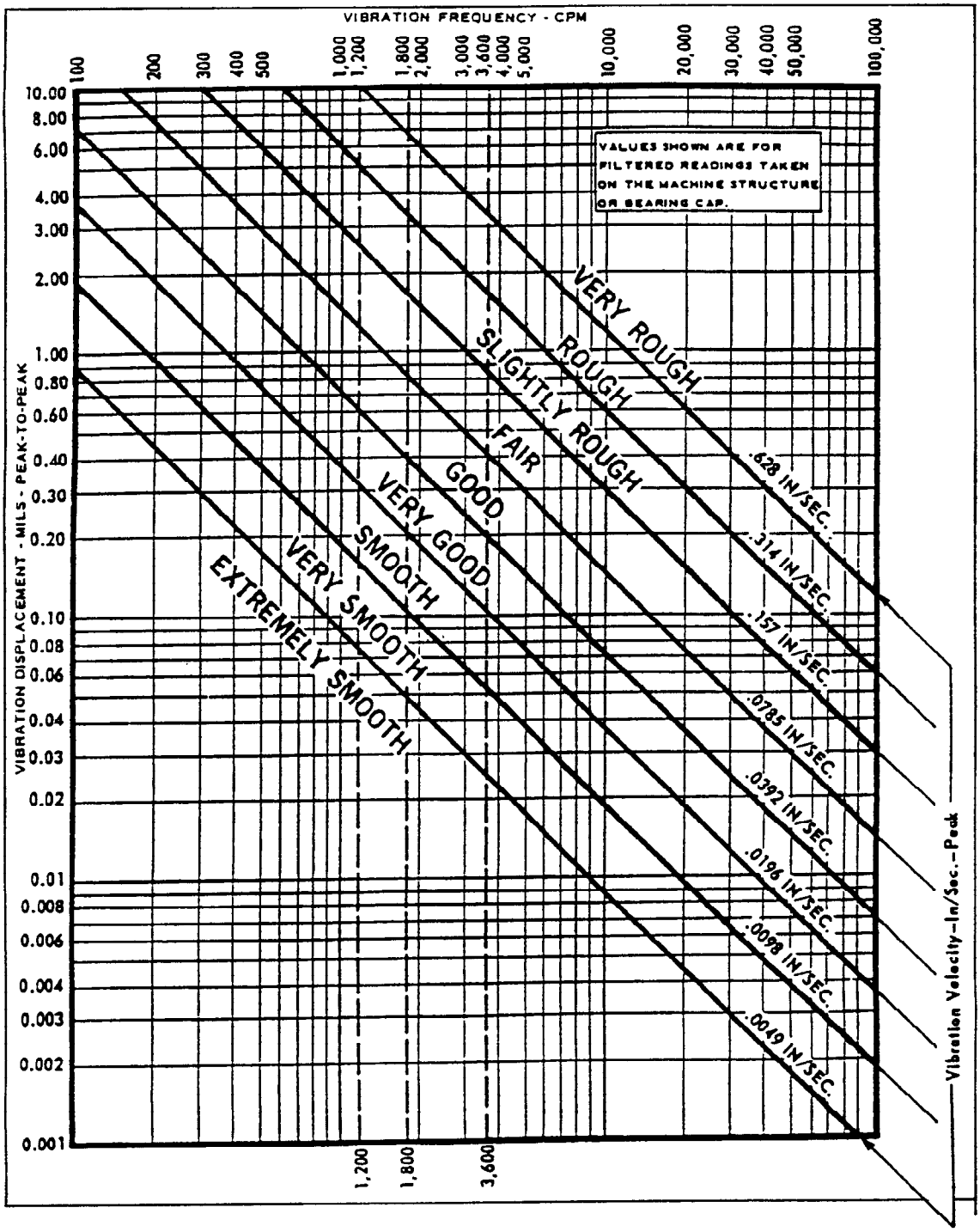


FIGURE 20
VIBRATION DISPLACEMENT & VELOCITY CHART FOR GENERAL HORIZONTAL ROTATING MACHINERY
(Source: Entek IRD International, Milford, OH)

© Copyright 2001 Technical Associates Of Charlotte, P.C.

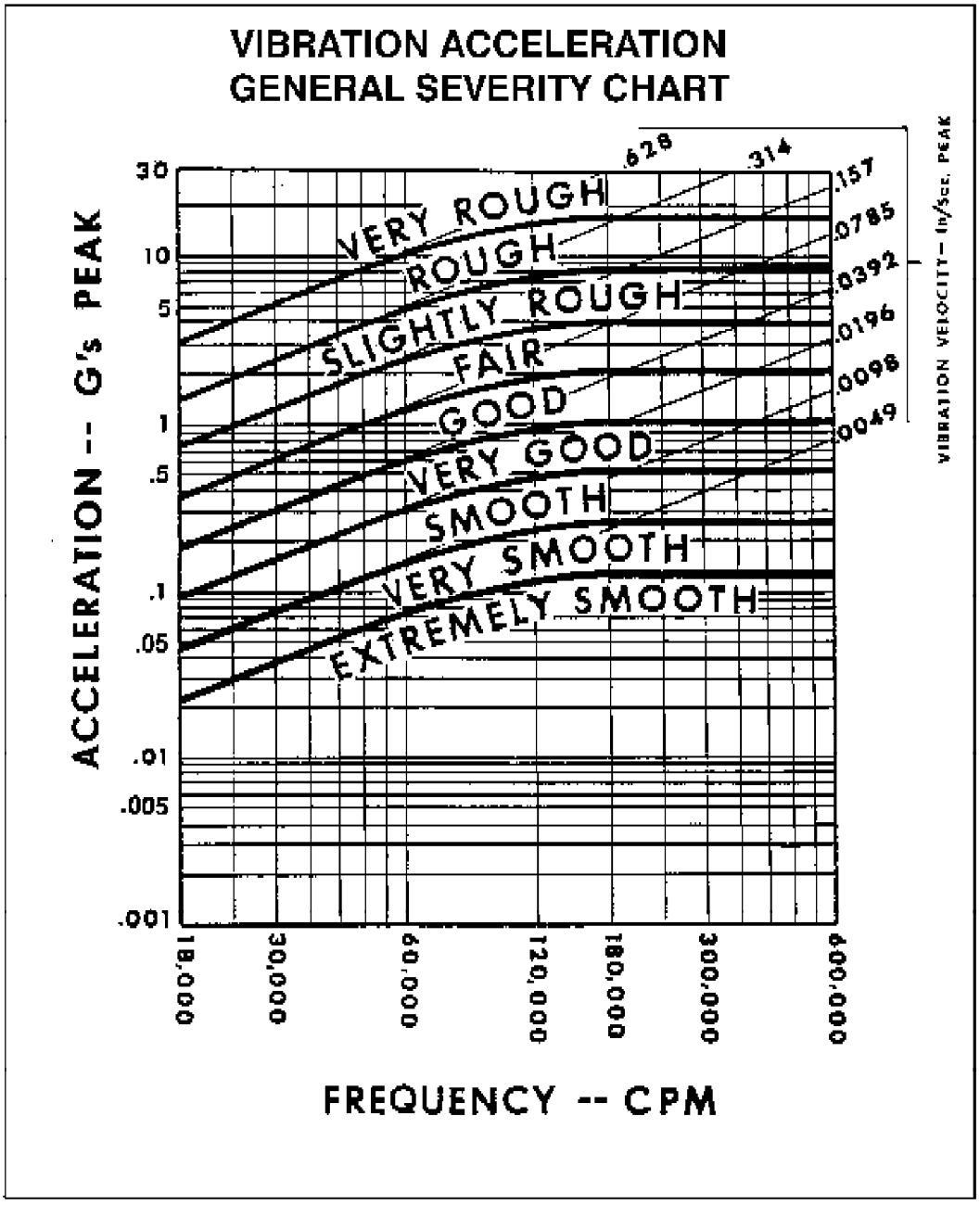


FIGURE 21
VIBRATION ACCELERATION & VELOCITY SEVERITY CHART FOR GENERAL HORIZONTAL ROTATING MACHINERY
(Source: Entek IRD International, Milford, OH)

© Copyright 2001 Technical Associates Of Charlotte, P.C.

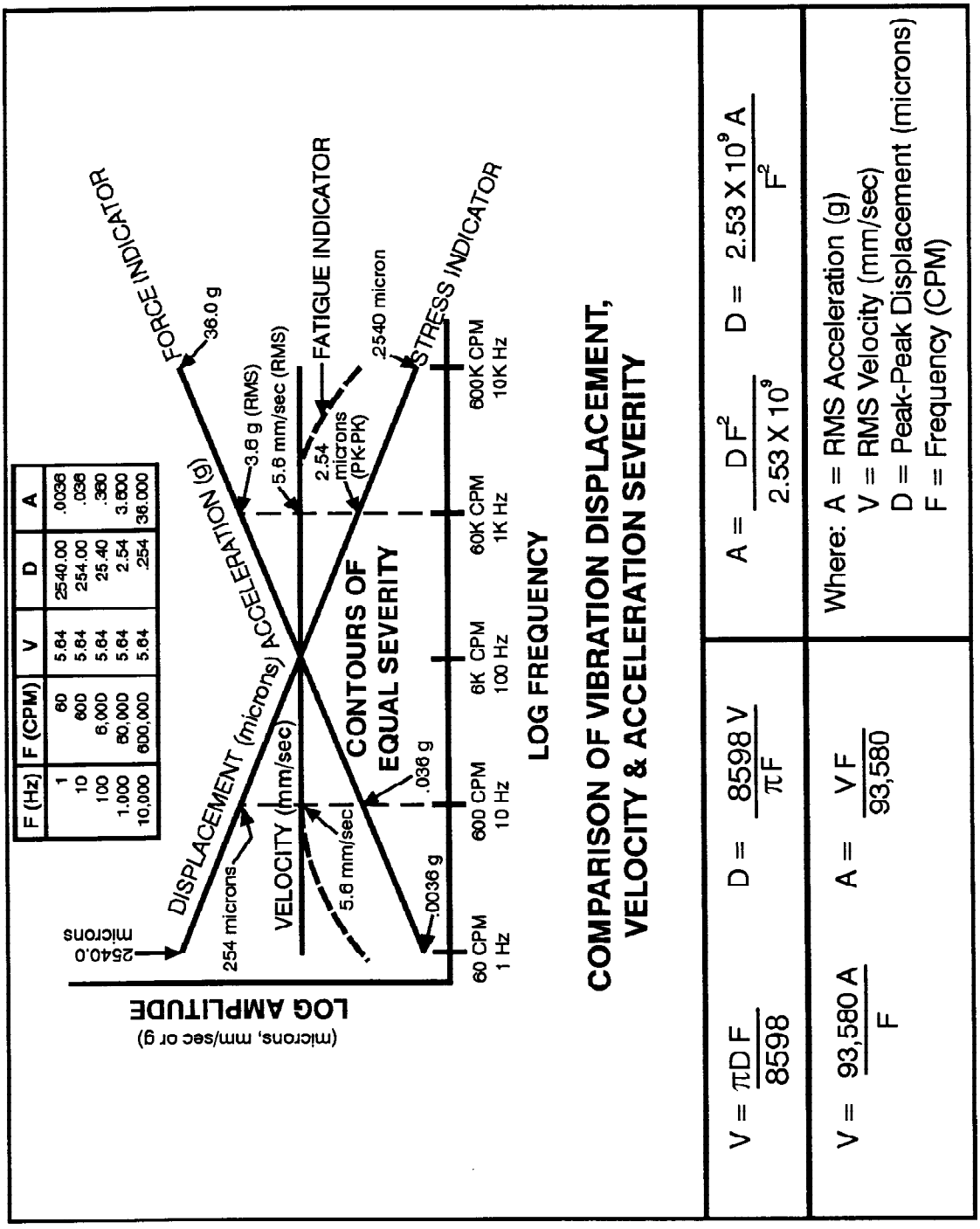


FIGURE 22
COMPARISON OF VIBRATION
DISPLACEMENT, VELOCITY & ACCELERATION

© Copyright 2001 Technical Associates Of Charlotte, P.C.

EXAMPLE A: CONVERSION OF VELOCITY TO DISPLACEMENT

(Peak in/sec to Peak-Peak mils)

$$V = \frac{\pi D F}{60{,}000} \qquad D = \frac{60{,}000 V}{\pi F}$$

Where:

V = Peak Velocity (in/sec.)
D = Peak-Peak Displacement (mils)
F = Frequency (CPM)

EXAMPLES:

① D = 2 mils Pk-Pk at 1800 CPM; V = ?

$$V = \frac{(\pi)(2)(1800)}{60{,}000}$$

V = .188 in/sec Peak

② V = .300 in/sec at 3600 CPM; D = ?

$$D = \frac{(60{,}000)(.300)}{(\pi)(3600)}$$

D = 1.59 mils Pk-Pk

③ D = .75 mil at 30,000 CPM; V = ?

$$V = \frac{(\pi)(.75)(30{,}000)}{(60{,}000)}$$

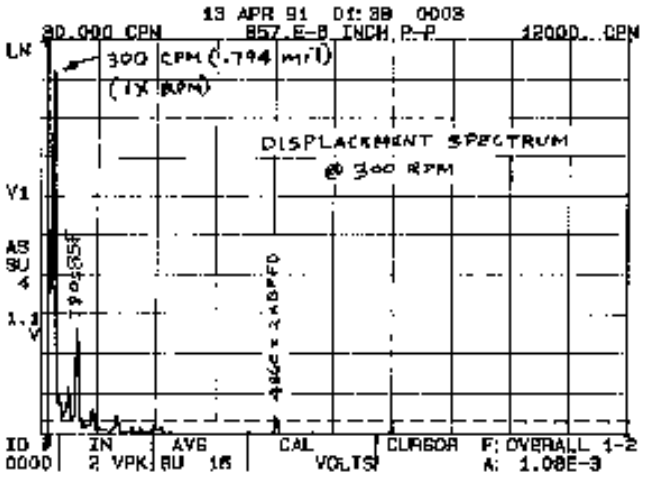
V = 1.18 in/sec Pk

④ V = .20 in/sec at 100 CPM; D = ?

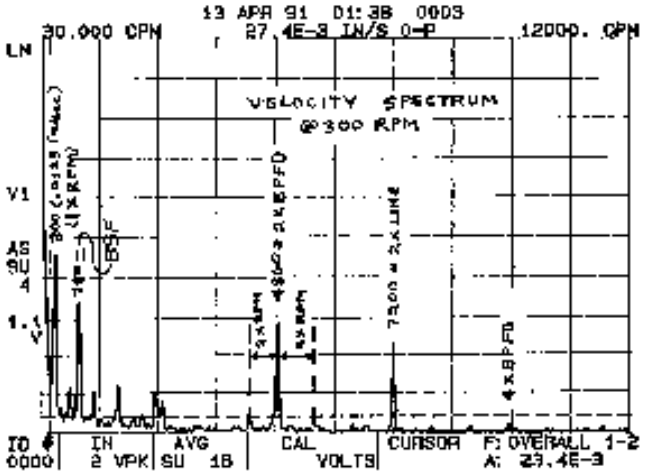
$$D = \frac{(60{,}000)(.20)}{(\pi)(100)}$$

D = 38.2 mils Pk-Pk

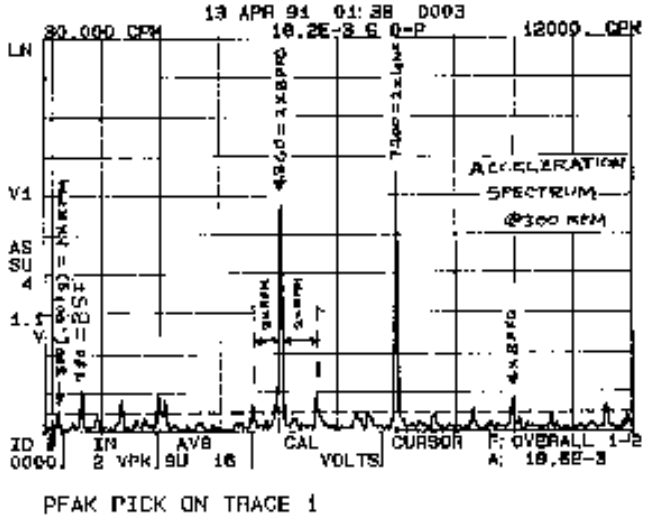
© Copyright 2001 Technical Associates Of Charlotte, P.C.



INCH P-P	CPM
794.E-6	300.00
82.2E-6	420.00
97.8E-6	600.00
227.E-6	780.00
50.9E-6	1110.0
39.1E-6	1589.9
31.3E-6	4860.0



IN/S 0-P	CPM
12.5E-3	300.00
1.75E-3	420.00
3.13E-3	600.00
9.26E-3	780.00
1.13E-3	900.00
2.88E-3	1110.0
1.13E-3	1500.0
3.38E-3	1589.9
1.13E-3	1890.0
1.25E-3	2070.0



G 0-P	CPM
997.E-6	300.00
1.91E-3	780.00
1.50E-3	1589.9
1.75E-3	2340.0
1.50E-3	2490.0
1.33E-3	4260.0
10.5E-3	4860.0
1.83E-3	5579.7
914.E-6	6360.0
914.E-6	6629.7
11.9E-3	7200.0

FIGURE 23
COMPARISON OF DISPLACEMENT, VELOCITY AND ACCELERATION SPECTRA ON A 300 RPM FAN WITH BEARING PROBLEMS

© Copyright 2001 Technical Associates Of Charlotte, P.C.

Therefore, it is important to note that a velocity spectrum has a wider usable frequency range for almost all rotating machinery than do either displacement or acceleration. Combining this characteristic with velocity's direct relationship with vibration severity almost always makes velocity the best vibration measurement unit to use (particularly when frequencies are below 120,000 CPM, or 2000 Hz).

Through the years, the general vibration severity chart of Figure 20 has probably been the most commonly used vibration severity chart for rotating machinery. Using this chart and velocity amplitudes, one can begin to assess how bad the vibration might be on his machine. However, this chart was never meant to apply to all machine types and configurations when choosing vibration limits that will give adequate warning of an existing or impending problem. To help meet this need, Technical Associates has also developed a more comprehensive vibration severity chart shown in Figure 24 which is entitled "CRITERIA FOR OVERALL CONDITION RATING". This chart applies to a wide variety of machines over a wide range of operating speeds from 600 to 60,000 RPM. It is important to point out these levels are peak overall velocity levels (in/sec). Note that two columns entitled "GOOD" and "FAIR" are included with the chart. These columns are used to give a machine an "Overall Condition Rating" based on the highest overall vibration level found on any of the machine measurement points.

However, these ratings are only applied if spectral band alarms specified by the user for these machines are not exceeded (in effect, these spectral alarms are bands which are placed around individual frequency peaks or groups of peaks and allow the user to specify a much higher alarm for 1X RPM than he would for other frequencies such as bearing defect frequencies). The information given in the condition rating chart of Figure 24 was acquired through many years of actual vibration data acquisition on a diverse array of machine types. Notice that an alarm value for a cooling tower fan drive outfitted with a long, hollow drive shaft (.600 in/sec) or a reciprocating compressor (.500 in/sec) is much higher than that allowed for a hermetic centrifugal chiller (.225 in/sec) or a machine tool motor (.175 in/sec). Also note that two alarm levels are given in the table (ALARM 1 and ALARM 2). In general, machines allowed to operate above Alarm 1 will likely fail prematurely if problems are not identified and corrected; while those allowed to operate above Alarm 2 may suffer catastrophic failure if left unaddressed.

The Technical Associates rating chart by no means covers all types of machines. For machines not included in the chart of Figure 24, one could use the Figure 20 severity chart or a statistical method to develop other alarm levels. The statistical method is especially effective when several identical machines are present, several surveys on the machines have been conducted and the machines' vibration levels can be compared to establish an acceptable vibration level. Using statistics to make this comparison lends a powerful mathematical approach. A statistical comparison can be conducted if the machines are similar in construction, drive configuration (direct coupled, belt driven, etc.), operating speeds, loading and internal components (bearings, gears and gear tooth count, etc.). If a statistical method is used, it is wise to revise the alarms as originally higher vibration levels are reduced through a vibration program.

Shock Pulse, HFD and Spike Energy are still another vibration measurement of special interest. Their purpose is to measure "bursts" of energy that occur in the ultrasonic frequency ranges and give very early indications of things such as minute surface flaws in bearings and gears. Until very recently these measurements were only viewed as an abstract numeric value. However, with new innovations in instrument circuitry these ultrasonic measurements are now able to be "demodulated" producing a meaningful spectrum. These ultrasonic measurements will be discussed in greater detail in another chapter.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

Metric Version Of Table II

TABLE II

TECHNICAL ASSOCIATES OF CHARLOTTE, P.C.

CRITERIA FOR OVERALL CONDITION RATING (RMS OVERALL VELOCITY, mm/sec.)*

1. Assuming Machine Speed = 600 to 60,000 RPM.
2. Assuming Measurements by Accelerometer or Velocity Pickup securely mounted as Close as Possible to Bearing Housing.
3. Assuming Machine Is Not Mounted on Vibration Isolators (for Isolated Machinery, Set Alarm 30% - 50% Higher).
4. Set Motor Alarms the Same as that for the Particular Machine Type unless Otherwise Noted.
5. Set Alarms on Individual External Gearbox Positions about 25% Higher than that for a particular Machine Type.

MACHINE TYPE	GOOD	FAIR	(WARNING) ALARM 1	(FAULT) ALARM 2
COOLING TOWER DRIVES				
Long, Hollow Drive Shaft	0 - 7.0	7.0 - 11.0	11.0	16.0
Close Coupled Belt Drive	0 - 5.0	5.0 - 7.5	7.5	12.0
Close Coupled Direct Drive	0 - 3.5	3.5 - 5.5	5.5	8.0
COMPRESSORS				
Reciprocating	0 - 6.0	6.0 - 9.0	9.0	13.5
Rotary Screw	0 - 5.5	5.5 - 8.0	8.0	12.0
Centrifugal With or W/O External Gearbox	0 - 3.5	3.5 - 5.5	5.5	8.0
Centrifugal - Integral Gear (Axial Meas.)	0 - 3.5	3.5 - 5.5	5.5	8.0
Centrifugal - Integral Gear (Radial Meas.)	0 - 2.5	2.5 - 4.5	4.5	6.5
BLOWERS (FANS)				
Lobe-Type Rotary	0 - 5.5	5.5 - 8.0	8.0	12.0
Belt-Driven Blowers	0 - 5.0	5.0 - 7.5	7.5	11.5
General Direct Drive Fans (with Coupling)	0 - 4.5	4.5 - 6.5	6.5	10.0
Primary Air Fans	0 - 4.5	4.5 - 6.5	6.5	10.0
Large Forced Draft Fans	0 - 3.5	3.5 - 5.5	5.5	8.0
Large Induced Draft Fans	0 - 3.0	3.0 - 5.0	5.0	7.0
Shaft-Mounted Integral Fan (Extended Motor Shaft)	0 - 3.0	3.0 - 5.0	5.0	7.0
Vane-Axial Fans	0 - 2.5	2.5 - 4.5	4.5	6.5
MOTOR/GENERATOR SETS				
Belt-Driven	0 - 5.0	5.0 - 7.5	7.5	12.0
Direct Coupled	0 - 3.5	3.5 - 5.5	5.5	8.0
CHILLERS				
Reciprocating	0 - 4.5	4.5 - 7.0	7.0	10.5
Centrifugal (Open-Air) - Motor & Comp. Separate	0 - 3.5	3.5 - 5.5	5.5	8.0
Centrifugal (Hermetic) - Motor & Impellers Inside	0 - 2.5	2.5 - 4.0	4.0	6.0
LARGE TURBINE/GENERATORS				
3600 RPM Turbine/Generators	0 - 3.0	3.0 - 5.0	5.0	7.0
1800 RPM Turbine/Generators	0 - 2.5	2.5 - 4.0	4.0	6.0
CENTRIFUGAL PUMPS				
Vertical Pumps (4m - 6m Height) } Height from Grade to Top Motor Bearing.	0 - 6.0	6.0 - 9.0	9.0	13.5
Vertical Pumps (3m - 4m Height) } May be Necessary to Spec. Lower Alarm	0 - 5.0	5.0 - 7.5	7.5	11.5
Vertical Pumps (2m - 3m Height) } for Lower Motor Bearing & for Upper	0 - 4.0	4.0 - 6.0	6.0	9.5
Vertical Pumps (0m -2m Height) } Pump Bearing (depending on height).	0 - 3.5	3.5 - 5.5	5.5	8.0
General Purpose Horizontal Pump - Direct Coupled	0 - 3.5	3.5 - 5.5	5.5	8.0
Boiler Feed Pumps - Horizontal Orientation	0 - 3.5	3.5 - 5.5	5.5	8.0
Hydraulic Pumps - Horizontal Orientation	0 - 2.0	2.0 - 3.5	3.5	5.5
MACHINE TOOLS				
Motor	0 - 1.2	1.2 - 1.8	1.8	2.7
Gearbox Input	0 - 1.8	1.8 - 2.7	2.7	4.0
Gearbox Output	0 - 1.2	1.2 - 1.8	1.8	2.7
Spindles:				
a. Roughing Operations	0 - .70	.70 - 1.2	1.2	1.8
b. Machine Finishing	0 - .45	.45 - .72	.72	1.1
c. Critical Finishing	0 - .27	.27 - .45	.45	.72

***NOTE: The "ALARM 1" and "ALARM 2" overall levels given above apply only to in-service machinery which has been operating for some time after initial installation and/or overhaul. They do not apply (and are not meant to serve as) Acceptance Criteria for either new or rebuilt machinery.**

COPYRIGHT© 1995 - TECHNICAL ASSOCIATES OF CHARLOTTE, P.C. **R=1097-3RMSMM**

FIGURE 24
CRITERIA FOR OVERALL CONDITION RATING
(PEAK OVERALL VELOCITY, IN/SEC)

2.5 UNDERSTANDING A VIBRATION SPECTRUM

A spectrum is a graphic display of vibration in the frequency domain with vibration amplitude (displacement, velocity or acceleration) on the Y-axis and frequency on the X-axis (CPM or Hz). Figure 25 is an example of a spectrum. However, note there is no way to tell how much or at what frequency the vibration is. Figure 26 is the same spectrum with units of measurement and other identifying information added. Note that the X-axis is delineated in CPM with a maximum frequency (F_{MAX}) of 12,000 CPM and the Y- axis is delineated in in/sec of vibration with .050 in/sec peak/division (with .050 in/sec per division X 10 divisions, this gives a Full Scale Amplitude of .500 in/sec). In this spectrum, both the horizontal and vertical scales have ten divisions with each frequency division being one-tenth of 12,000 CPM. Note the other identifying information presented on the spectrum that includes the rotational speed, machine name (TEST 12K400L), the measurement position (4H), the date and time the data was taken and the units of measurement (in/sec).

The large peak about 1.5 divisions over in the spectrum is the dominant vibration. It would be possible to scale the amplitude and frequency values by using a measuring device such as a ruler but that would only be a close estimate. However, the addition of a cursor as shown in Figure 27 makes determining frequency and amplitude of an individual peak easy. In Figure 27, the cursor is at the dominant peak and indicates this vibration is .2474 in/sec at 1770 RPM (the rotating speed of the machine). Addition of an amplitude threshold level of 1% places an imaginary line across the spectrum at 1% of the full amplitude scale of .500 in/sec as shown in Figure 27. What results from the threshold is a listing of all the peaks that exceed the 1% level (.005 in/sec in this case), their order (order = multiple of running speed) and their amplitude (in/sec in this case). Please note that each vibration software package will present the information in Figure 27 using its own format.

2.51 EFFECT ON FREQUENCY ACCURACY OF # FFT LINES USED

It is very important that both amplitude and frequency values are known as precisely as possible for all peaks in the spectrum when doing an analysis. Amplitude of course is important as it gives one an idea of the severity of the problem. Frequency on the other hand is used to determine the source of the vibration. For instance the dominant vibration, (vibration with the largest peak) in the spectrum of Figure 27 is at 1770 CPM. The rotational speed of the machine (1X RPM) is also 1770 RPM. Therefore, the component's rotating speed at 1770 RPM is the source of the dominant vibration. Knowing that the problem source is at 1X RPM helps one know a list of possible problem sources, and at the same time, eliminates the possibility of other sources being the problem such as rolling element bearing, blade pass, or most all electrical problems.

Probably the greatest requirement in determining potential machine problem source(s) is knowing the actual value of the frequency as accurately as possible. Determining an accurate vibration frequency can become clouded or enhanced depending on how many lines of spectral resolution are used in collecting and displaying the data. Most modern computer monitors are able to display 400 lines of spectral resolution (and most data collectors today are usually set up to capture 400 line spectra on their PMP routes). That is, a spectral display is made of 400 individual vertical lines (or "bins") located adjacent to one another along the frequency axis. Each one of these bins has amplitude information stored in it relative to the amount of vibration at that specific frequency or bin location. Not all bins contain information since the vibration at some frequency bin locations is zero. Thus in Figure 27, the bin that contains the 1770 CPM frequency is the 59th of 400 bins as counted from the left (or zero) end of the frequency scale. Looking at Figure 27, the 400th bin containing 12,000 CPM (12K CPM) has zero amplitude indicating no vibration is present at that frequency. In the case of Figure 27, each bin (line of resolution) covers a range of

© Copyright 2001 Technical Associates Of Charlotte, P.C.

frequencies. The frequency range per bin (known as frequency resolution) is calculated by dividing F_{MAX} (12,000 CPM) by the number of lines of resolution (400 lines of resolution in this case) or 12,000 CPM/400 lines of resolution = 30 CPM/line of resolution (bin). Therefore each bin has a 30 CPM range. The size of this resolution has a definite effect on the frequency information gained from a spectrum. For instance, the 1770 CPM bin of Figure 27 actually contains vibration information in the range of 1755 to 1785 CPM. So whether there truly are one or more peaks within this range is accumulatively represented by the 1770 CPM frequency and .2474 in/sec amplitude (i.e., if a vibration existed at 1760 CPM in addition to the 1770 CPM vibration, the amplitude of the 1760 CPM frequency would contribute to the total amplitude shown for the 1770 CPM bin.

The higher the lines of resolution specified for a spectrum, the more precise will be the frequency read by the analyzer and displayed in the spectrum. Note importantly that the precision of any displayed frequency will be the frequency reading plus or minus one-half the resolution. That is, if a 12,000 CPM spectrum with 30 CPM frequency resolution were captured and showed a peak of 1800 CPM, the accuracy would be 1800 CPM (±15 CPM), or from 1785 to 1815 CPM. On the other hand, if a 120,000 CPM spectrum using 400 lines were taken (300 CPM resolution) at the same location and showed the same 1800 CPM peak, the accuracy would be ± 150 CPM or from 1650 to 1950 CPM. Figures 27 through 31 are spectra with 100, 200, 400 and 3200 lines of resolution. Note in each spectrum how the vibration peaks are displayed. The 100 line resolution spectrum of Figure 28 has a very “blocked” appearance with the dominant vibration shown as being 1800 CPM. With 100 lines of resolution available, each bin has a frequency range of 120 CPM/line of resolution included. Therefore, considerable inaccuracy is seen in determining the actual vibration frequency. Figure 29 has 200 lines of resolution. Therefore, its resolution is slightly better than that of Figure 28 with 60 CPM resolution. Here we see the dominant vibration is said to be at 1740 CPM. Figure 30 is a 400 line spectrum with 30 CPM/line of resolution accuracy and identifies the peak at 1770 CPM, a much more accurate vibration representation than the 100 or 200 line spectra. Further accuracy can be achieved by using higher numbers of lines of resolution. Figure 31 is a portion of a 3200 line resolution spectrum known as a “zoom spectrum”. Calculating the resolution of the 3200 line spectrum yields 3.75 CPM/line of resolution accuracy for the 12,000 CPM frequency span spectrum. Thus, looking at the 3200 line spectra of Figure 31, the dominant vibration is shown as 1758.7 CPM (and is truly therefore 1758.7 ± 1.87 CPM). This degree of accuracy is not always needed, but is extremely helpful when vibration frequencies from different machine sources are very close to each other. For example, bearing defect frequencies are commonly close to exact multiples of operating speed. Consider a pump impeller with 6 vanes and bearings which have a defect frequency of say 6.03X RPM. In this case, it may require a high resolution spectrum to separate the bearing frequency from what is known as the blade (or vane) passing frequency which equals the number of vanes X RPM (6X RPM here).

So why shouldn’t one always take 3200 line resolution spectra and avoid the hassle of inaccurate data? Collecting 3200 line data is costly in time as well as in data collector and computer storage. First, a 3200 line spectrum takes 8 times longer to collect than does a 400 line spectrum as seen by Equation 2 and can really slow down the data collection process. For example, for a frequency span of 12,000 CPM (200 Hz) and a frequency resolution of 400 lines, the required data acquisition time (t_{MAX}) would be 60 X 400/12,000 = 2 sec. If eight (8) averages were desired, a total of 16 seconds would be required just to capture the data - not including either analyzer settling time nor the time to perform the mathematical calculation of the FFT (assuming no overlap processing is used in the analyzer).

© Copyright 2001 Technical Associates Of Charlotte, P.C.

Second, some data collectors and PMP software currently only allow 3200 lines of resolution in a special analysis mode, and not during regular route data collection. For most general data collection 400 lines of resolution produces adequate accuracy unless one happens to have sources producing frequencies located close to one another such as the above example pump. Some recent software packages are now permitting 3200 line spectra for route data collection, but judgement must be used since the data collector spectral storage capacity will be significantly reduced for such high resolution spectra.

$$\text{Data Sampling Time} = \frac{(60)\,(\#\ \text{FFT Lines})\,(\#\text{Avg.})}{\text{Freq. Span}} \qquad \text{Equation 2}$$

Where:

Data Sampling Time	=	Total Sampling Time Required to Capture Data (sec.); (Assuming no Overlap Processing is involved).
# FFT Lines	=	Number of # FFT Lines or Bins in Spectrum.
# Avg.	=	Number of Averages.
Freq. Span	=	Frequency Span (CPM).

2.52 EFFECT ON FREQUENCY ACCURANCY OF FREQUENCY SPAN USED

Another factor affecting resolution is the frequency range (F_{MAX}) setting. The larger the frequency range, the less accurate will be the frequency reading and the wider will be each line (bin) of resolution. Figure 32, is a 400 line spectrum with a 24,000 CPM (24K CPM) F_{MAX}. The resolution of this spectrum is 24,000 CPM/400 lines of resolution = 60 CPM/line of resolution. Thus, its frequency definition is only one-half that of the 12,000 CPM (12K CPM) F_{MAX} 400 line spectrum of Figure 30. Further, Figure 33 is a 120,000 CPM F_{MAX} spectrum with 400 lines of resolution. Here, Figure 33 has 300 CPM per line of resolution meaning that the accuracy of any displayed frequency will be ± 150 CPM (or plus or minus one-half the resolution). In some cases, this amount of accuracy is all that is needed. Again, trade-offs must be made to retain accuracy. The analyst needs to have a large enough F_{MAX} to include all the vibration data he needs, but doesn't want to lose accuracy in the process. Technical Associates has a proven method for choosing F_{MAX} values for data collection. The method is discussed in detail in another paper.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

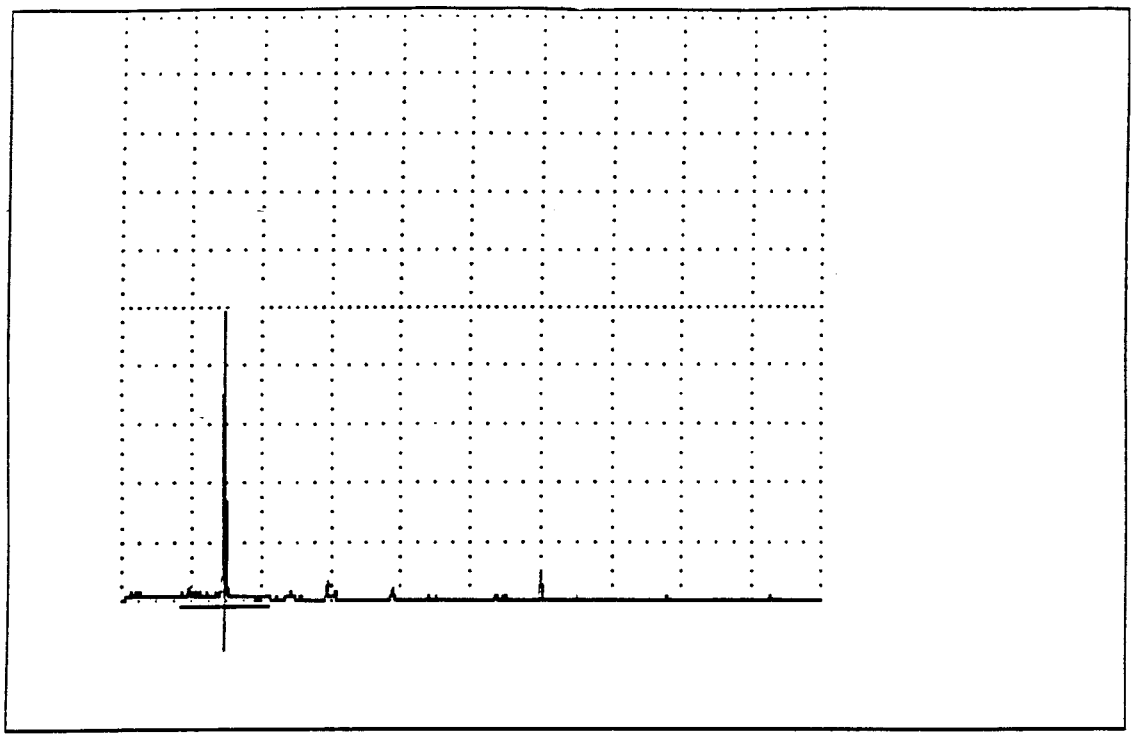


FIGURE 25
EXAMPLE VIBRATION SPECTRUM

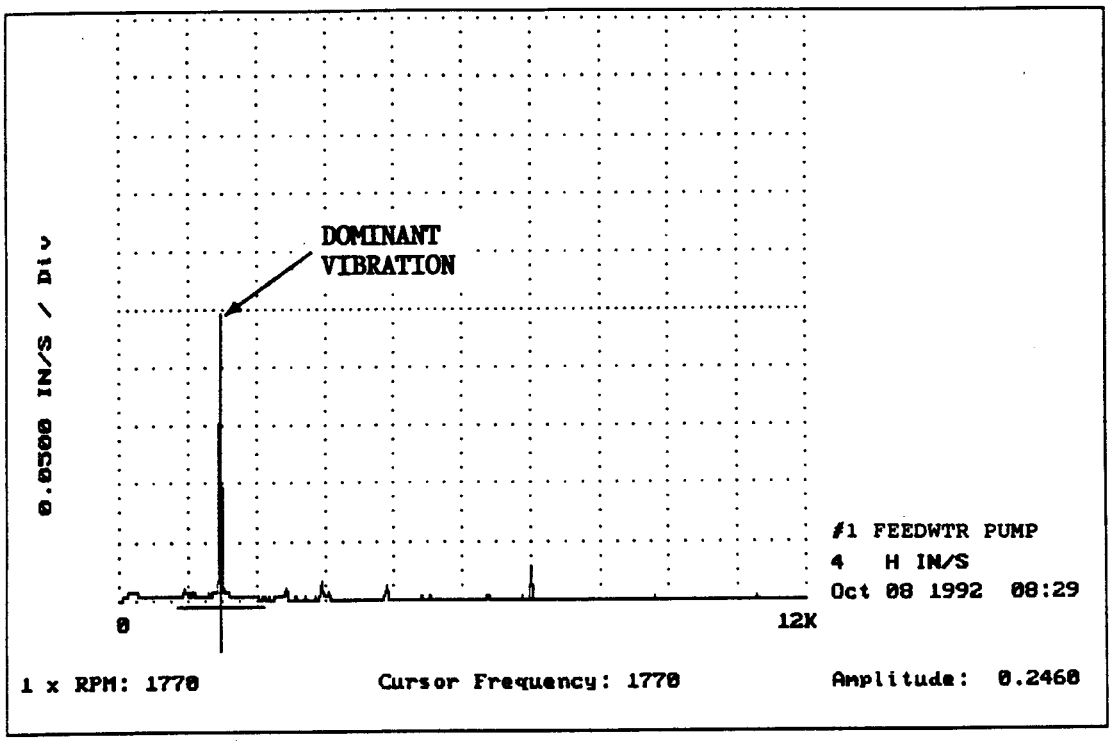
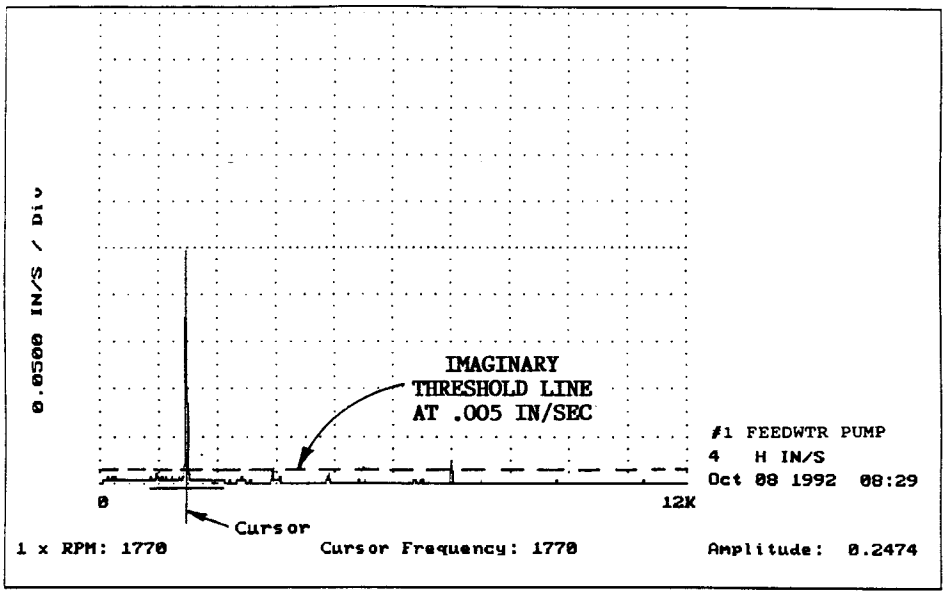


FIGURE 26
SPECTRUM WITH UNITS OF MEASUREMENT
AND OTHER INFORMATION ADDED

© Copyright 2001 Technical Associates Of Charlotte, P.C.



1% x 0.500 = 0.0050 1 x RPM: 1770.00

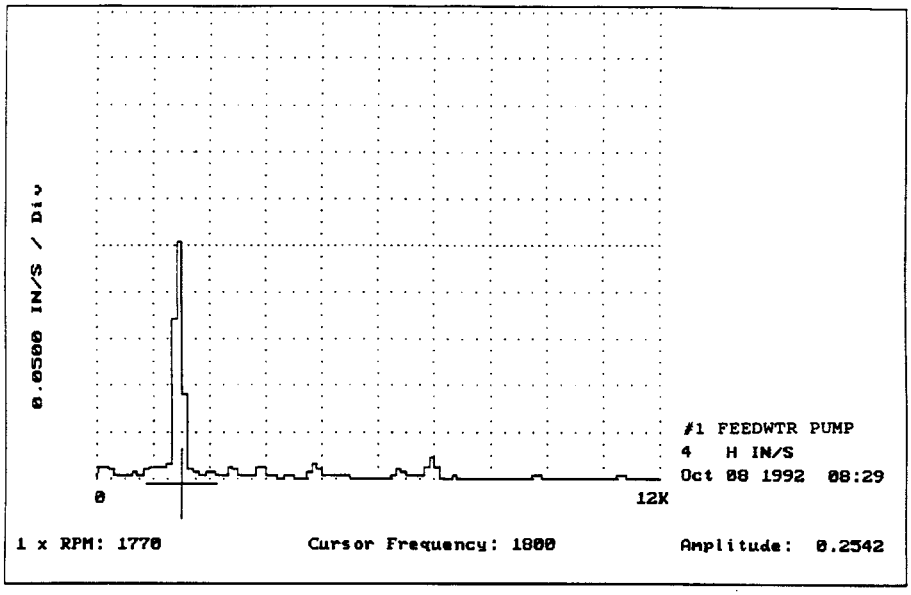
Machine	Pos	Dir	Units	Date
#1 FEEDWTR PUMP	4	H	IN/S	Oct 08 1992 08:29

Frequency	1 x RPM	Amplitude
180	0.102	0.0054
240	0.136	0.0057
300	0.169	0.0070
1050	0.593	0.0062
1170	0.661	0.0088
1260	0.712	0.0053
1350	0.763	0.0053
1470	0.831	0.0053
1620	0.915	0.0051
1770	1.000	0.2474
2910	1.644	0.0078
3540	2.000	0.0144
3660	2.068	0.0055
4680	2.644	0.0108
7200	4.068	0.0234

T - Threshold adjust Q - Quit

FIGURE 27
SPECTRUM WITH CURSOR AND FREQUENCY SCAN PRINTOUT

© Copyright 2001 Technical Associates Of Charlotte, P.C.



001% x 0.500 = 0.0050 1 x RPM: 1770.00

Machine	Pos	Dir	Units	Date
#1 FEEDWTR PUMP	4	H	IN/S	Oct 08 1992 08:2

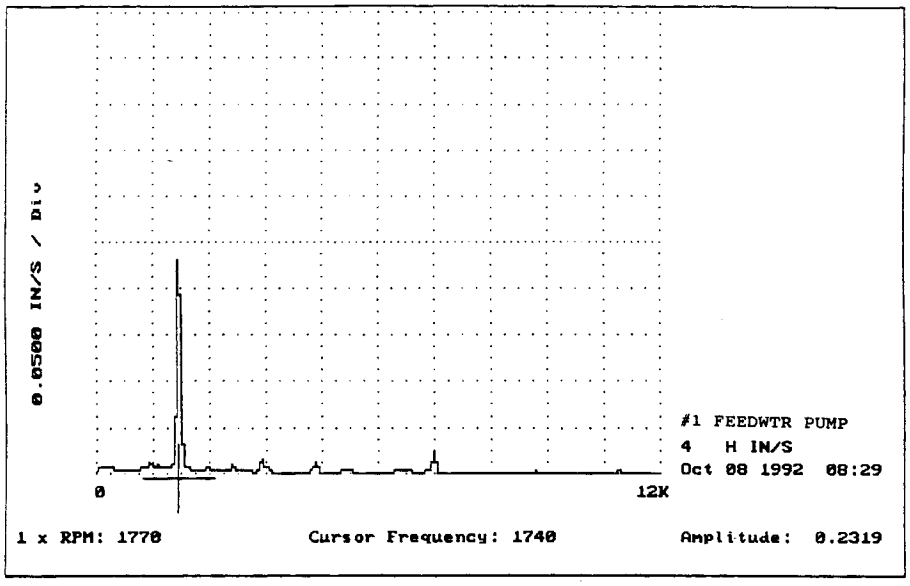
Frequency	1 x RPM	Amplitude
240	0.136	0.0129
840	0.475	0.0053
1200	0.678	0.0132
1800	1.017	0.2542
2400	1.356	0.0061
2880	1.627	0.0115
3600	2.034	0.0134
4680	2.644	0.0154
6480	3.661	0.0081
7200	4.068	0.0227

$$\text{FREQ. RESOLUTION} = \frac{12{,}000 \text{ CPM}}{100 \text{ LINES}} = 120 \text{ CPM/LINE}$$

T - Threshold adjust Q - Quit

FIGURE 28
12,000 CPM SPECTRUM WITH 100 LINES OF RESOLUTION

© Copyright 2001 Technical Associates Of Charlotte, P.C.



1% x 0.500 = 0.0050 1 x RPM: 1770.00

Machine	Pos	Dir	Units	Date
#1 FEEDWTR PUMP	4	H	IN/S	Oct 08 1992 08.29

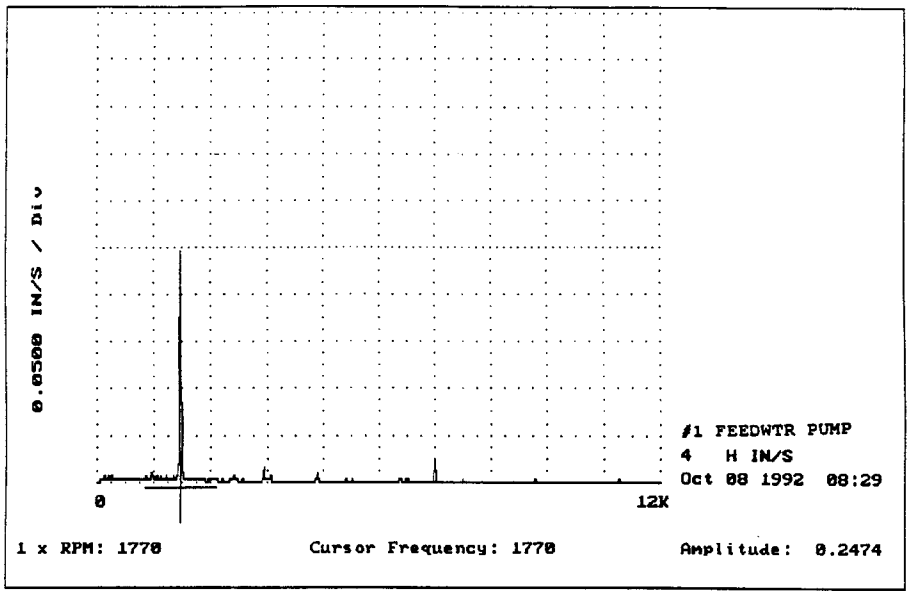
Frequency	1 x RPM	Amplitude
120	0.068	0.0076
300	0.169	0.0072
1140	0.644	0.0111
1320	0.746	0.0087
1500	0.847	0.0075
1740	0.983	0.2319
2400	1.356	0.0051
2880	1.627	0.0083
3540	2.000	0.0160
4680	2.644	0.0116
7200	4.068	0.0237

$$\text{FREQ. RESOLUTION} = \frac{12{,}000 \text{ CPM}}{200 \text{ LINES}} = 60 \text{ CPM/LINE}$$

T - Threshold adjust Q - Quit

FIGURE 29
12,000 CPM SPECTRUM WITH 200 LINES OF RESOLUTION

© Copyright 2001 Technical Associates Of Charlotte, P.C.



1% x 0.500 = 0.0050 1 x RPM: 1770.00

Machine	Pos	Dir	Units	Date
#1 FEEDWTR PUMP	4	H	IN/S	Oct 08 1992 08:29

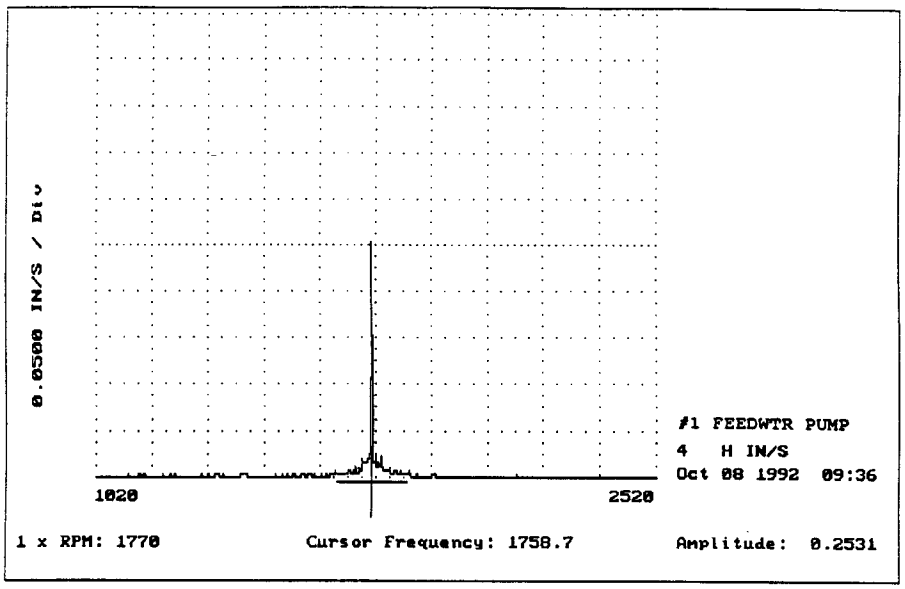
Frequency	1 x RPM	Amplitude
180	0.102	0.0054
240	0.136	0.0057
300	0.169	0.0070
1050	0.593	0.0062
1170	0.661	0.0088
1260	0.712	0.0053
1350	0.763	0.0053
1470	0.831	0.0053
1620	0.915	0.0051
1770	1.000	0.2474
2910	1.644	0.0078
3540	2.000	0.0144
3660	2.068	0.0055
4680	2.644	0.0108
7200	4.068	0.0234

$$\text{FREQ. RESOLUTION} = \frac{12{,}000 \text{ CPM}}{400 \text{ LINES}} = 30 \text{ CPM/LINE}$$

T - Threshold adjust Q - Quit

FIGURE 30
12,000 CPM SPECTRUM WITH 400 LINES OF RESOLUTION

© Copyright 2001 Technical Associates Of Charlotte, P.C.



006% x 0.500 = 0.0300 1 x RPM: 1770.00

Machine	Pos	Dir	Units	Date
#1 FEEDWTR PUMP	4	H	IN/S	Oct 08 1992 09:36

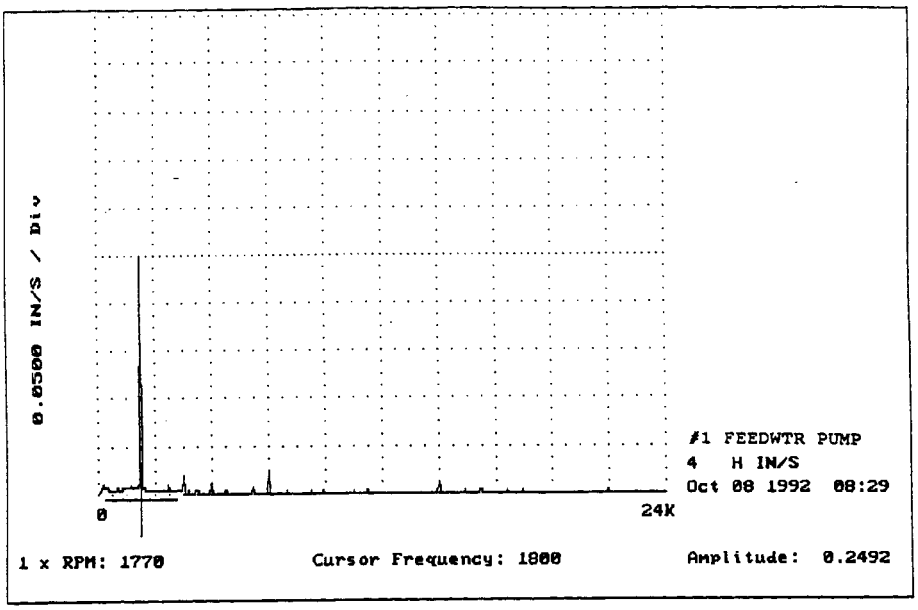
Frequency	1 x RPM	Amplitude
1758.7	0.994	0.2531

FREQ. RESOLUTION = $\frac{12{,}000 \text{ CPM}}{3200 \text{ LINES}}$ = 3.75 CPM/LINE

T - Threshold adjust Q - Quit

FIGURE 31
ZOOM SPECTRA TAKEN FROM A 12,000 CPM FREQUENCY SPAN SPECTRUM HAVING 3200 LINES OF RESOLUTION

© Copyright 2001 Technical Associates Of Charlotte, P.C.



1% x 0.500 = 0.0050 1 x RPM: 1770.00

Machine	Pos	Dir	Units	Date
#1 FEEDWTR PUMP	4	H	IN/S	Oct 08 1992 08:2

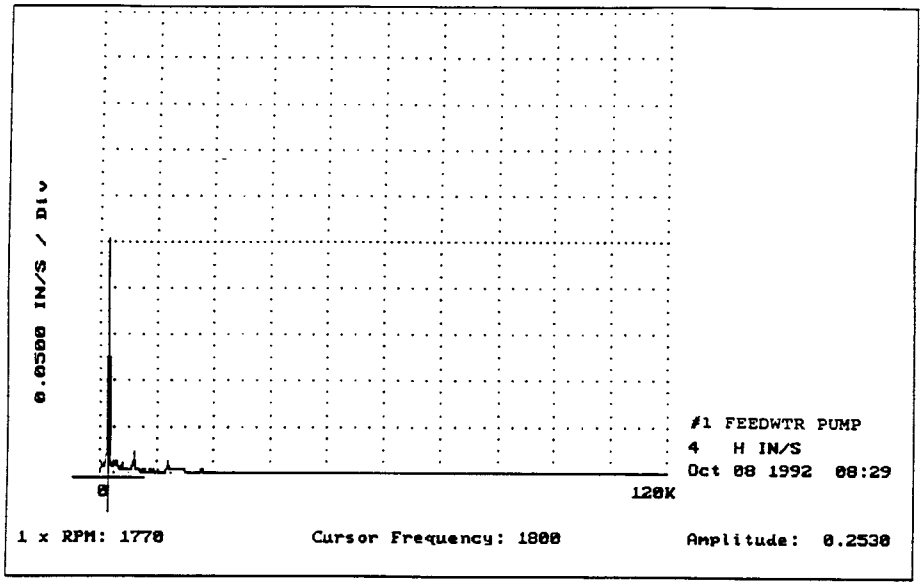
Frequency	1 x RPM	Amplitude
180	0.102	0.0095
780	0.441	0.0050
1020	0.576	0.0066
1140	0.644	0.0076
1380	0.780	0.0097
1500	0.847	0.0078
1800	1.017	0.2492
2940	1.661	0.0083
3600	2.034	0.0178
4740	2.678	0.0105
7200	4.068	0.0253
14400	8.136	0.0111

FREQ. RESOLUTION = $\frac{24{,}000 \text{ CPM}}{400 \text{ LINES}}$ = 60 CPM/LINE

T - Threshold adjust Q - Quit

FIGURE 32
24,000 CPM SPECTRUM WITH 400 LINES OF RESOLUTION

© Copyright 2001 Technical Associates Of Charlotte, P.C.



1% x 0.500 = 0.0050 1 x RPM: 1770.00

Machine	Pos	Dir	Units	Date
#1 FEEDWTR PUMP	4	H	IN/S	Oct 08 1992 08:29

Frequency	1 x RPM	Amplitude
300	0.17	0.0098
1800	1.02	0.2530
3000	1.69	0.0117
3600	2.03	0.0137
4800	2.71	0.0091
7200	4.07	0.0230
14400	8.14	0.0111

$$\text{FREQ. RESOLUTION} = \frac{120{,}000 \text{ CPM}}{400 \text{ LINES}} = 300 \text{ CPM/LINE}$$

T - Threshold adjust Q - Quit

FIGURE 33
120,000 CPM SPECTRUM WITH 400 LINES OF RESOLUTION

© Copyright 2001 Technical Associates Of Charlotte, P.C.

2.6 WHAT IS OVERALL VIBRATION (DIGITAL AND ANALOG OVERALL LEVEL)?

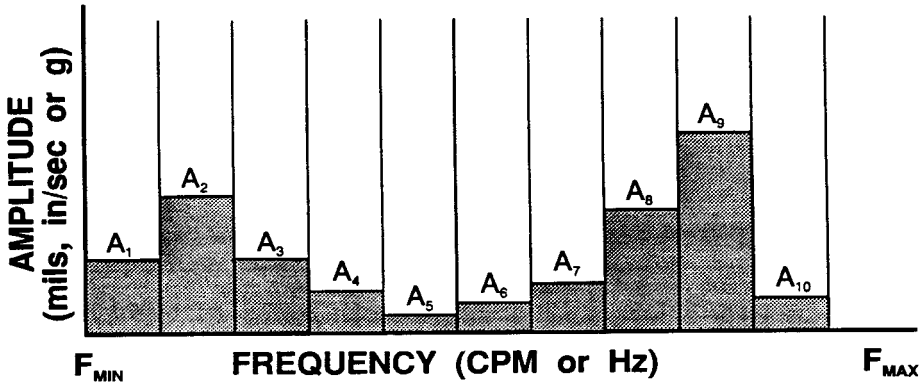
The vibration seen in a spectrum is the total sum of all the vibration measured by the transducer within the frequency span chosen. The transducer may even sense vibration from an adjacent machine and include it within the vibration spectrum. Whatever is seen by the transducer, no matter what its source or from where it might originate, becomes part of both the spectrum and the overall vibration. Overall vibration is different from vibration at specific frequencies as discussed thus far in that it is a total summation of all the vibration, no matter what the frequency. Figure 34 shows the overall vibration level of a spectrum (this is sometimes called the Digital Overall. Simply said, if you take the amplitude of each frequency bin (Ai) and square it (A_i^2); add all the squared amplitudes together; take the square root of that sum; and divide this sum by the noise factor for the FFT Window chosen (the Hanning window is most always used for PMP route measurements and has a 1.5 noise factor), the end result is the overall vibration of the spectrum. Of course, this is a very lengthy process for a 400 line spectrum (and even more so for a 3200 line FFT). However, this spectral overall is automatically calculated either within the analyzer and/or back within the host software of the computer.

Figures 35 and 36 show an approximate formula for calculating the overall vibration and on accompanying sample spectrum showing how this approximation is applied. Here, the amplitudes from 5 separate frequency peaks were combined to compute the approximate overall level of the spectrum. Figure 36 is a sample estimated overall vibration calculation using an actual spectrum. Note the estimated overall vibration calculated is .161 in/sec. The actual overall as measured by the data collector was .182 in/sec. The actual overall should always be higher than the estimated method as it includes all the individual amplitudes within each bin, not just the detectable peaks over the user-defined threshold. Multiplying the estimated value by a 1.1 correction factor will usually make the estimated vibration fairly close to the actual (i.e., 1.1 X .161 = .177 in/sec, very close to the actual .182 in/sec overall vibration). Fortunately, the data collectors and computer software perform all these calculations and one only needs to do this calculation under unusual circumstances.

One of the problems with looking at only the Spectrum Overall is that significant vibration can occur outside the frequency range (0 - F_{MAX}) specified by the analyst. For example, if a high level of .60 in/sec were occurring out at a frequency of 100,000 CPM and an F_{MAX} of only 60,000 CPM were chosen, this high amplitude 100,000 CPM frequency would not be included within the Spectral Overall. In this case, the Spectral Overall might calculate up to only .20 in/sec, far below that if a frequency range out to 120,000 CPM had been chosen.

To overcome this problem, some PMP systems determine the overall by looking directly at the time waveform which itself has a very wide frequency range. Then, the overall measured by the analyzer will be totally independent of any frequency span spectrum chosen by the user. This is what is known as the Analog Overall. For example, one system uses a time waveform having a fixed frequency span from 300 to 3,900,000 CPM (5 to 65,000 Hz). Then, in the example given above with the .60 in/sec at 100,000 CPM, the Analog Overall would be dramatically higher than the Spectrum (or Digital) Overall if an F_{MAX} of only 60,000 CPM were chosen. For this reason if your PMP system allows you the option of finding either the Spectrum (or Digital) Overall or Analog Overall, choose the Analog Overall option.

© Copyright 2001 Technical Associates Of Charlotte, P.C.



$$OA_{RSS} = \frac{\sqrt{\sum_{i=1}^{n} A_i^2}}{\sqrt{N_{BF}}} = \frac{\sqrt{A_1^2+A_2^2+A_3^2+....+A_n^2}}{\sqrt{1.5}}$$

OA_{RSS} = Digital Overall Level of Vibration Spectrum.
RSS = Root Sum Square Level which is the "Summation of Energy" in the spectrum from F_{MIN} to F_{MAX}.
n = Number of FFT Lines of Resolution.
A_i = Amplitude of Each of the FFT Lines.
N_{BF} = Noise Bandwidth for Window Chosen 1.5 for Hanning Window.

Which means $OA_{RSS} = .8165\sqrt{\sum_{i=1}^{n} A_i^2}$

FIGURE 34
EXACT EQUATION FOR CALCULATING DIGITAL OVERALL LEVEL OF A SPECTRUM

© Copyright 2001 Technical Associates Of Charlotte, P.C.

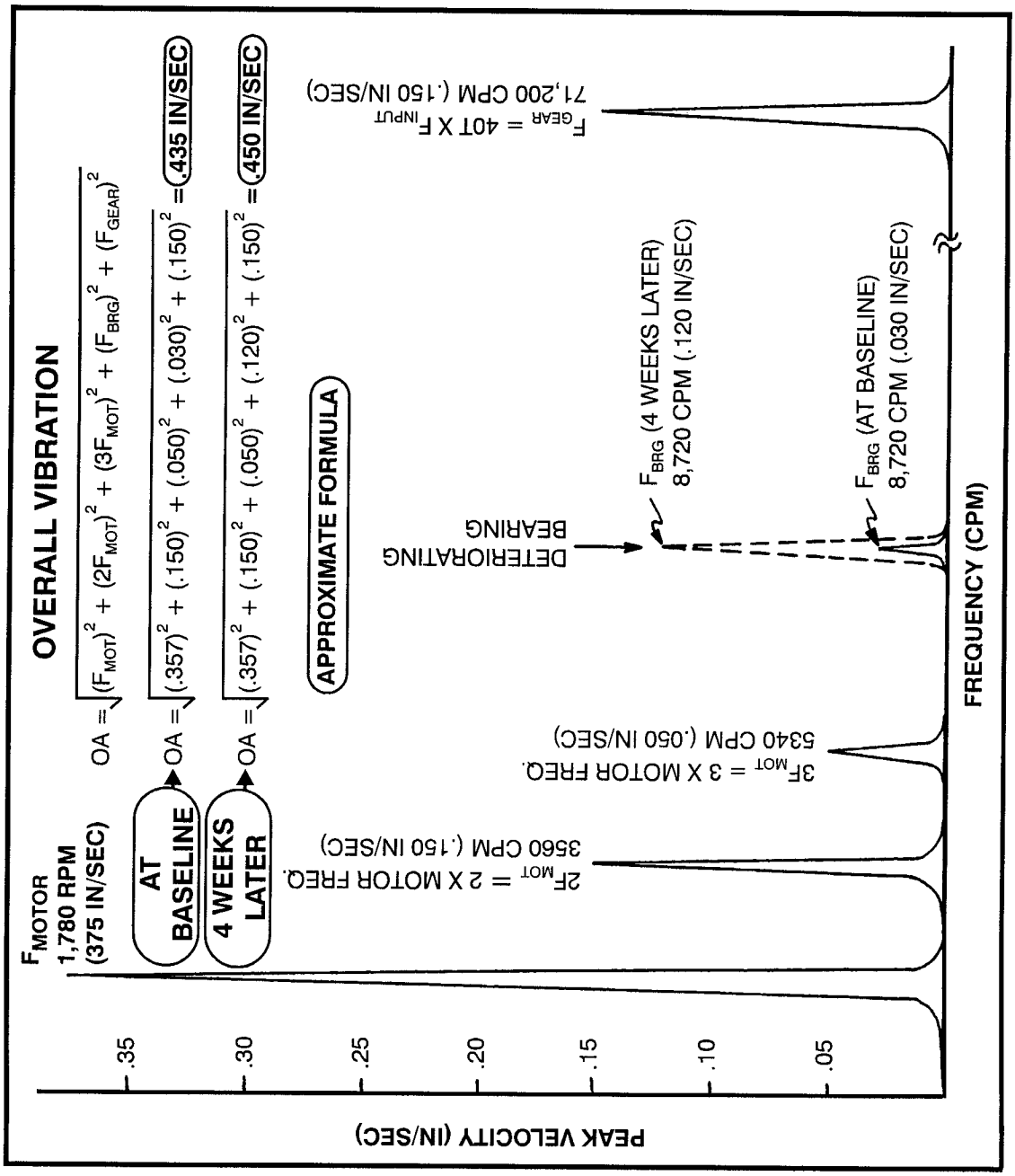
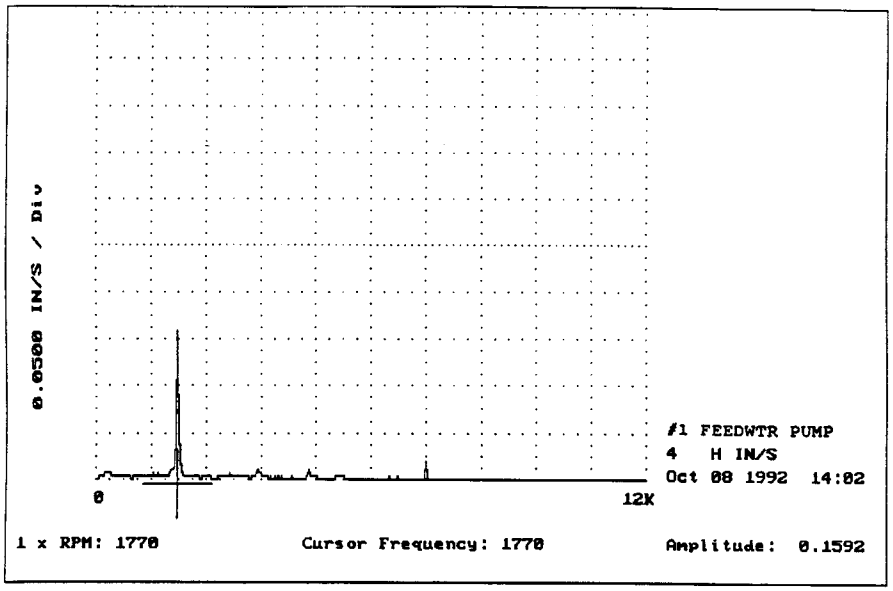


FIGURE 35
APPROXIMATE FORMULA FOR CALCULATING DIGITAL OVERALL LEVEL OF A SPECTRUM

© Copyright 2001 Technical Associates Of Charlotte, P.C.



1% x 0.500 = 0.0050 1 x RPM: 1770.00

Machine	Pos	Dir	Units	Date
#1 FEEDWTR PUMP	4	H	IN/S	Oct 08 1992 14:02

Frequency	1 x RPM	Amplitude
240	0.136	0.0057
300	0.169	0.0071
1230	0.695	0.0051
1770	1.000	0.1592
2910	1.644	0.0050
3510	1.983	0.0106
4650	2.627	0.0079
7200	4.068	0.0201

where OA = Overall Level of Vibration
n = Number of FFT Lines of Resolution
Ai = Amplitude of Each of the FFT Lines

$$OA = \sqrt{\sum_{i=1}^{n} A_i^2}$$

$$OA = \sqrt{(A_1)^2 + (A_2)^2 + (A_3)^2 + (A_4)^2 + (A_5)^2 + (A_6)^2 + (A_7)^2 + (A_8)^2}$$

$$OA = \sqrt{(.0057)^2+(.0071)^2+(.0051)^2+(.1592)^2+(.0050)^2+(.0106)^2+(.0079)^2+(.0201)^2}$$

$$= \sqrt{.026}$$

OA = .161 inch/sec
OA = .161 X 1.1 (CORRECTION FACTOR
OA = .177 inch/sec

T - Threshold adjust Q - Quit

FIGURE 36
SAMPLE CALCULATION OF APPROXIMATE OVERALL (OA) VIBRATION

© Copyright 2001 Technical Associates Of Charlotte, P.C.

2.7 UNDERSTANDING PHASE AND ITS APPLICATIONS

2.71 DEFINITION OF PHASE:

Phase analysis is a powerful tool one can employ to diagnose dominant problem sources. For example, there are many problems which can cause high vibration at operating speed (i.e., unbalance, misalignment, eccentricity, bent shaft, soft foot, cracked/broken gear tooth, resonance, loose hold-down bolts, etc.). Likewise, there are several problems which can generate high vibration at either 2X or 3X RPM. With all these possible problems potentially capable of generating high vibration at these frequencies, an analyst can be hard pressed to determine with conviction the dominant source(s). However, when high levels do occur at 1X, 2X and/or 3X RPM, phase measurements at each bearing housing in question can go far in doing just this - if the analyst clearly understands phase.

Phase is the relationship vibration has with respect to another vibrating part or fixed reference point (it can also be thought of as the vibration motion at one location relative to the vibration motion of another location). It can also be thought of as the timing relationship between two signals occurring at the same frequency. Phase is easiest to visualize if one is familiar with using a timing light to set the timing of an automobile engine. The timing light is actuated by the spark going to a certain spark plug (usually the “Number 1” plug). Here, the object is to have the spark synchronized with the position of the “Number 1” piston for proper firing. This is done by adjusting the spark timing so the timing mark on the crankshaft aligns (synchronizes) with a fixed reference mark on the engine block. This assures the spark firing and piston positions are synchronized.

Vibration phase is very similar to the above example except that the vibration (not the spark) is the trigger, and the timing of the vibration is measured by its angular position relative to the vibration trigger instead of being adjusted to suit a predetermined timing setup. Strobe lights and photoelectric cells (using reflective tape) are both used to measure phase.

2.72 HOW TO MAKE PHASE MEASUREMENTS:

Phase measurements are a powerful tool in diagnosing machinery vibration. Figures 37 and 38 illustrate a typical strobe light method for taking phase measurements.

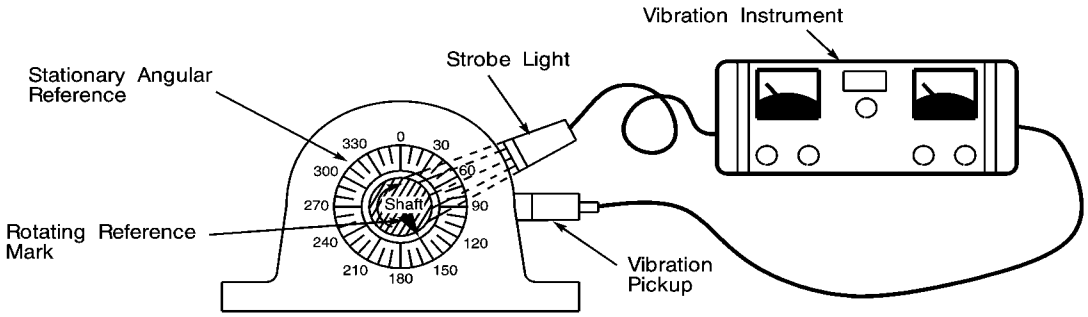


FIGURE 37
PHASE REFERENCE WITH A ROTATING REFERENCE MARK AND A STATIONARY ANGULAR REFERENCE

© Copyright 2001 Technical Associates Of Charlotte, P.C.

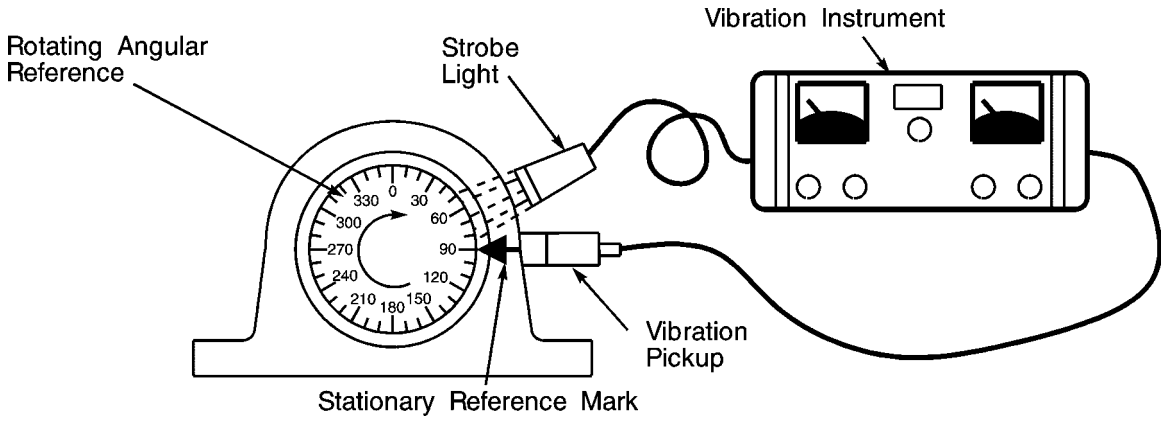


FIGURE 38
PHASE MEASUREMENTS WITH A STATIONARY REFERENCE MARK AND ROTATING ANGULAR REFERENCE

The photocell method shown in Figure 39 utilizes a stationary photocell targeted at a piece of reflective tape mounted to the rotating component of interest.

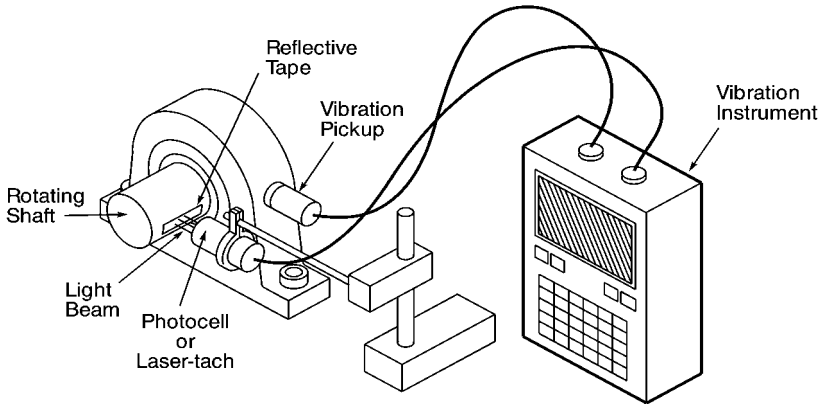


FIGURE 39
PHOTOCELL METHOD OF ACQUIRING PHASE MEASUREMENTS

The photocell method is more accurate than either hand-held strobe light method since the instrument measures the phase angle within very accurate tolerances, whereas the strobe methods include human error in attempting to accurately read the angular position of the reference mark. However, since the strobe method is more visual, it will be helpful to continue our discussion using this method.

Importantly, to properly collect phase measurements, the strobe light must first be “tuned” to the frequency of interest (usually the speed of some rotating shaft) and the vibration amplitude and phase must be recorded. The transducer should be moved from one bearing to another taking data in horizontal, vertical and axial directions. In the case of a strobe light, one will only have to find and record the new position of the reference mark each time the transducer is moved to a new location or direction. He can manipulate the strobe light any way he likes to best see the reference mark, but the transducer must be firmly mounted and kept in place for each reading (however, if a photocell is being used, both the photocell and transducer will have to be locked down; with each subsequent measurement, only the transducer will be moved to the next location).

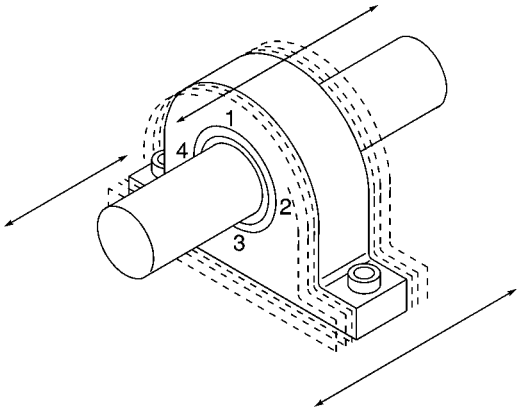
© Copyright 2001 Technical Associates Of Charlotte, P.C.

2.73 Using Phase Analysis in Vibration Diagnostics:

Here, what one is examining is whether the whole bearing face is moving back and forth directly (as in Figure 40), or if it is vibrating with a twisting motion as shown in Figure 41).

2.731 Evaluating Axial Motion of a Bearing Housing to Reveal a Possible Cocked Bearing or Bent Shaft:

Using the convention shown in Figure 40, the transducer should be mounted at locations 1, 2, 3 and 4 when taking axial phase data. Figure 40 indicates (as it should) planar motion of the bearing.



PHASE	
1	150°
2	150°
3	150°
4	150°

FIGURE 40
AXIAL PHASE SHOWING PLANAR MOTION

However, if the same four measurements produced results of Figure 41 with 90° phase differences at each location, a bent shaft or cocked bearing would be suspected. In this case, the 180° phase change across points 1 and 3 would indicate an up and down twist, while that between points 2 and 4 would show a side-to- side twist possibly caused by either a bent shaft or cocked bearing.

PHASE	
1	60°
2	150°
3	240°
4	330°

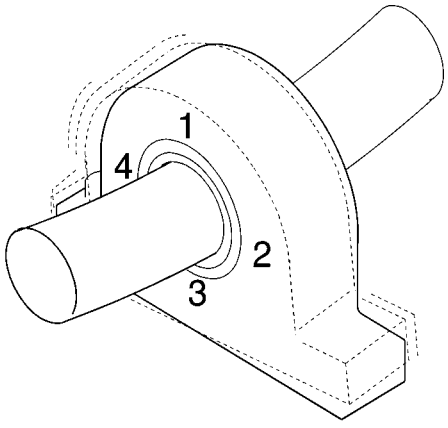


FIGURE 41
AXIAL PHASE SHOWING TWISTING MOTION DUE TO BENT SHAFT OR COCKED BEARING

© Copyright 2001 Technical Associates Of Charlotte, P.C.

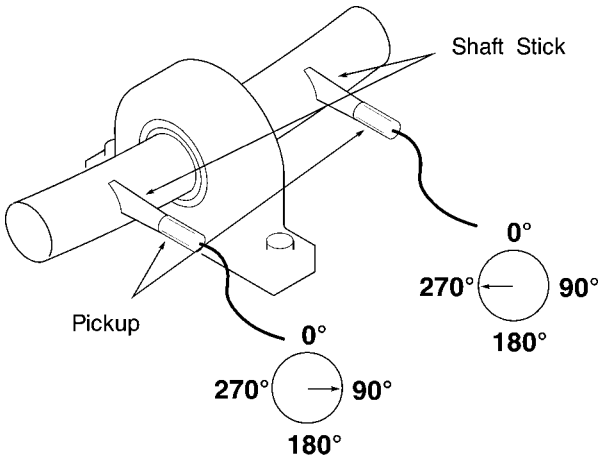


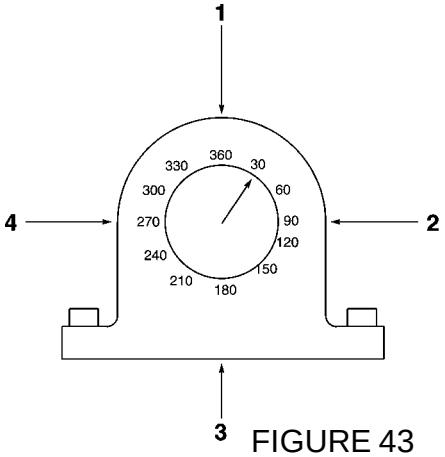
FIGURE 42
USING A SHAFT STICK TO CONFIRM A BENT SHAFT

Further phase analysis could be employed using a shaft stick on either side of the bearing to confirm a bent shaft as shown in Figure 42.

In this case, the phase would be 180° out-of-phase as long as the transducers themselves are pointing in the same direction. (Note: Reversal of direction of the transducer by 180° will automatically change the phase measurement by 180°!)

Unbalance is usually indicated by taking radial phase measurements as illustrated in Figure 43.

2.732 Phase Behavior Due to Unbalance:



PHASE	
1	30°
2	120°
3	210°
4	300°

FIGURE 43
RADIAL PHASE MEASUREMENTS FOR UNBALANCE

An unbalance condition will normally have 90° phase increments between each of tradimeasurement locations which are likewise 90° apart. Significant departure from this configuration means the problem is usually something other than unbalance. An even better indicator of unbalance is comparing phase measurements on both the outboard and inboard bearings of a rotor as shown in Figure 44.

Note in Figure 44 that one compares the horizontal and vertical phase motion on the outboard and inboard bearings to confirm whether or not significant unbalance exists. Do this by finding difference in phase in the horizontal direction on the

© Copyright 2001 Technical Associates Of Charlotte, P.C.

outboard and inboard bearings. Then find the vertical phase difference between measurements on the outboard and inboard bearings. If unbalance is significant, 1X RPM vibration must be high and the horizontal phase difference should equal the vertical phase difference between the outboard and inboard bearings (±30°). That is, this shows the resultant motion of the rotor is the same in both the horizontal and vertical directions (if not, the dominant problem is something other than unbalance). For example, in the dynamic unbalance example shown in Table C of Figure 44, note that the horizontal phase difference on the motor bearings (Positions 1 & 2) is 60° (90° - 30°) compared to a vertical phase difference of 60° (180° - 120°), which strongly points to unbalance.

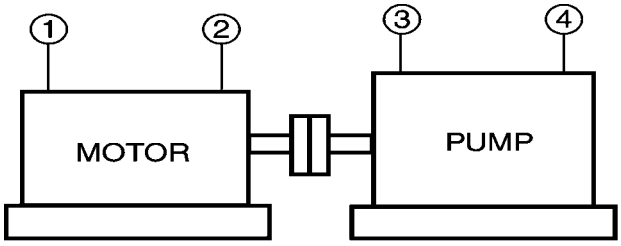


TABLE A.
PHASE AT 1X RPM FOR A MACHINE
WITH DOMINANT STATIC UNBALANCE

DIR.	1	2	3	4
A	60°	*70°	60°	*80°
H	30°	25°	30°	40°
V	120°	110°	120°	135°

* Corrected Phase Values Accounting For 180° Orientation of Accelerometer.

TABLE B.
PHASE AT 1X RPM FOR A MACHINE
WITH DOMINANT COUPLE UNBALANCE

DIR.	1	2	3	4
A	60°	*70°	60°	*80°
H	30°	210°	200°	220°
V	120°	295°	280°	300°

* Corrected Phase Values Accounting For 180° Orientation of Accelerometer.

TABLE C.
PHASE AT 1X RPM FOR A MACHINE
WITH SIGNIFICANT DYNAMIC UNBALANCE

DIR.	1	2	3	4
A	60°	*70°	60°	*80°
H	30°	90°	80°	70°
V	120°	180°	170°	165°

* Corrected Phase Values Accounting For 180° Orientation of Accelerometer.

FIGURE 44
TYPICAL PHASE MEASUREMENTS WHICH WOULD INDICATE EITHER STATIC, COUPLE OR DYNAMIC UNBALANCE

© Copyright 2001 Technical Associates Of Charlotte, P.C.

2.733 Phase Behavior Due to Looseness/Weakness:

Certain types of looseness are detectable with phase analysis. Tightly secured machine components should move in unison with one another having no significant phase or amplitude differences between the individual pieces. If the amplitude and/or phases change appreciably between the mated parts shown in Figure 45, looseness/ weakness should certainly be suspected. Looking at Figure 45, note the significant amplitude and phase change between measurements on the baseplate and the supporting concrete base. This indicates a problem which often might be caused by inadequate grouting between these two surfaces.

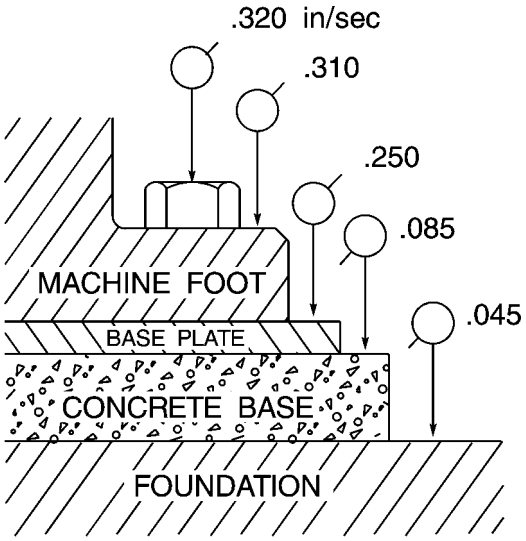


FIGURE 45
PHASE MEASUREMENTS FOR LOOSENESS

2.734 Phase Behavior Due to Misalignment:

Misalignment detection is probably one of the most common uses for employing phase analysis. Both parallel (offset) and angular shaft misalignment are detectable through phase measurements. Figure 46 illustrates the two types of shaft misalignment.

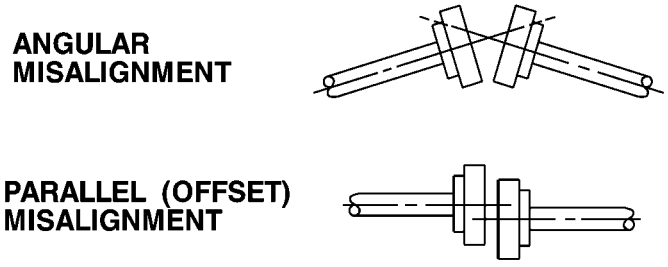


FIGURE 46
DIAGRAM OF ANGULAR AND PARALLEL (OFFSET)
SHAFT MISALIGNMENT

© Copyright 2001 Technical Associates Of Charlotte, P.C.

Shaft misalignment is characterized by phase differences approaching 180° across the coupling. Amplitude and phase measurements should be taken both axially and radially in the 4 quadrant locations on two adjacent bearings. To check coupling alignment, the two bearings need to be on either side of the coupling. Radial phase is most sensitive to parallel misalignment and axial phase to angular misalignment. In comparing phase measurements of adjacent bearings it is important that direction conventions are maintained for the transducer (i.e., reversing direction of the transducer will result in a 180° phase shift and can falsely indicate a misalignment). It is also important that the 4 quadrant positions be noted from the same viewing (or reference) direction as shown in Figure 47 to avoid corrupted phase data.

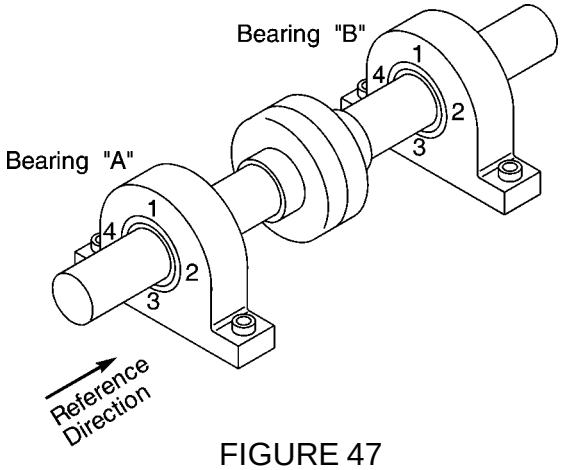
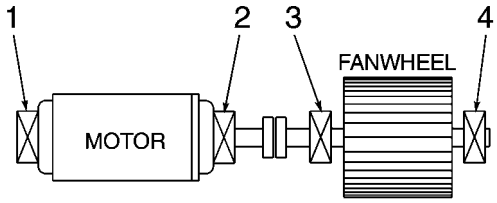


FIGURE 47
COORDINATING PHASE MEASUREMENT POSITIONS ON ADJACENT BEARINGS

In Figure 47 Bearing A phase measurements at Location 2 must be compared to the corresponding Location 2 of Bearing B. If the quadrant locations of Bearing B had been located from the backside of Bearing B, there is the danger of the numbering sequence progressing around the bearing in the opposite direction.

Figure 48 is an example of coupling misalignment. For simplicity, the measurements are only shown for the axial direction.



1	2	3	4
1 90°	1 90°	1 270°	1 270°
2 105°	2 105°	2 270°	2 255°
3 105°	3 90°	3 270°	3 270°
4 90°	4 85°	4 270°	4 270°

Note: Phase measurements have been corrected for transducer direction.

FIGURE 48
AXIAL PHASE COMPARISONS FOR ANGULAR MISALIGNMENT

© Copyright 2001 Technical Associates Of Charlotte, P.C.

From the data provided in Figure 48, motor Bearings 1 and 2 are moving in unison with one another, and Bearings 3 and 4 are also moving together indicating there is no internal misalignment, bent shaft or cocked bearings in either the motor or fan. However, the 180° phase change across the coupling indicates shaft misalignment between Bearings 2 and 3.

Note that most misalignment is a combination of both parallel and angular misalignment. Rarely do either types exist in a pure state. Also, note how the phase measurements are not exactly 0° or 180° different with respect to one another. Usually, phase measurements within 30° of each other are considered “approximately in phase”.

However, the more nearly the phase differences approach 180° , the higher the probability of misalignment, whether this significant phase change is either in the axial or radial direction.

2.735 Using Phase Analysis to Find the Operating Deflection Shape of a Machine and its Support Frame:

Phase measurements are also useful in determining the operating deflection shape of a machine or structure. By dividing the structure into several equally spaced measurement locations (usually 10 to 12), and taking phase measurements relative to the frequency of interest (1X RPM, 2X RPM, etc.), operating deflection of the machine and support structure can be determined. This can be a powerful tool in revealing potential problem sources such as soft foot, misalignment, resonance, etc.

Figure 49 shows some equally spaced phase measurement locations on a machine base that have an apparent flexing problem at 1X RPM of the machine mounted to it. The object is to determine the location of maximum and minimum flexure to explore possible methods of reducing excessive motions (whether the vibration is a problem at 1X RPM, 2X RPM, 6X RPM in the case of a 6-vaned pump impeller, etc.).

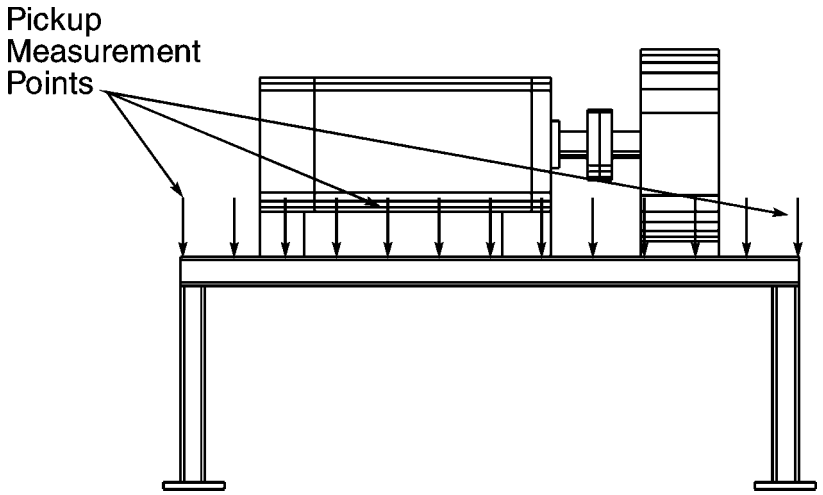


FIGURE 49
AMPLITUDE AND PHASE MEASUREMENT LOCATIONS ON FLEXING MACHINE BASE USED TO DETERMINE ITS OPERATING DEFLECTION SHAPE

© Copyright 2001 Technical Associates Of Charlotte, P.C.

Both vibration amplitude and phase need to be recorded at each of the measurement locations. The amplitude measurements can then be plotted on a drawing of the base structure as shown in Figure 50. Note that the length of each vertical line is drawn to be proportional to the vibration amplitude. Also, note that the amplitude lines (vectors) were all presumed to go upwards and that a node at each end as well as in the middle of the span is present.

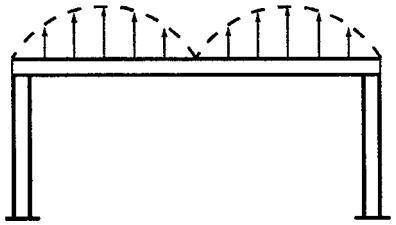


FIGURE 50
PLOTTING PROPORTIONAL LINES TO BUILD THE OPERATING DEFLECTION SHAPE WITHOUT PHASE MEASUREMENTS

Adding the phase information shows true deflection shape in Figure 51. Figure 51 shows the phase data actually indicated the left and right halves of the structure were moving in opposite directions. The nodes at center span and at the end points were unaffected, but the actual operating shape was.

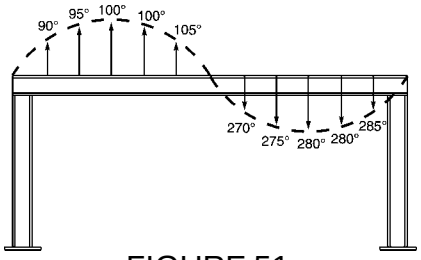


FIGURE 51
ADDITION OF PHASE TO THE OPERATING DEFLECTION SHAPE DRAWING

This type of structural analysis is very important as it shows stiffening at the center of the span as in Figure 52 would likely have little effect.

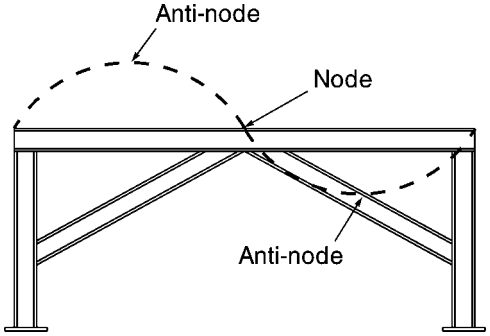


FIGURE 52
INEFFECTIVE APPLICATION OF STIFFENING AT A NODE

© Copyright 2001 Technical Associates Of Charlotte, P.C.

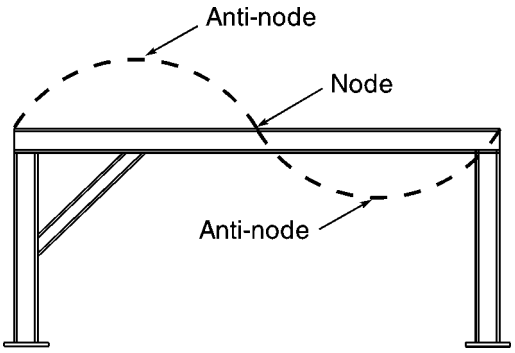


FIGURE 53
STIFFENING AT THE ANTI-NODE AS DETERMINED BY THE MODE SHAPE

The more effective location of such a brace would be at an anti-node point where vibration is the highest as shown in Figure 53 - that is, at a point 1/4 the span from either end (only one brace is required).

Operating deflection shape software is now available that will even animate the shape on a computer monitor. This makes the technique even more powerful, particularly when attempting to show the results of such evaluations to a person inexperienced in vibration analysis. Such software is highly recommended by the author.

REFERENCE:

1.) Vibration Technology - 1; Published by IRD Mechanalysis; Columbus, OH; 1989; Page 2-7.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

CHAPTER 3

OVERVIEW OF THE STRENGTHS AND WEAKNESSES OF TYPICAL VIBRATION INSTRUMENTS

3.0 INTRODUCTION

The purpose of vibration instrumentation is to accurately measure vibration amplitudes, frequencies, and phase so that a reliable determination of a machine's condition can be diagnosed. There are a whole array of vibration instruments having a wide range of capabilities. Basically, there are 5 different types of vibration measurement instruments:

1) Overall Level Vibration Meters (Hand-Held)

2) Swept-Filter Analyzers

3) FFT Data Collectors (also Hand-Held)

4) Real-Time Spectrum Analyzers

5) Instrument Quality Tape Recorders (both older Analog Types and newer Digital Types)

Each of these instrument types has its own optimum application in a vibration monitoring/ analyzing program. When deciding which type of instrument to purchase, or use in a particular vibration test, many characteristics of the instrument must be considered. The purpose of this chapter is to give the reader a basic understanding of these characteristics and to tabulate which instrument possesses which characteristics. Table I lists many of the items which must be considered when detecting which instrument to use or which to purchase. It is important to note that the following information is meant only to reflect the general properties and capabilities of instrument configurations. Not all makes and models of vibration measurement instruments will exhibit these characteristics. However, this section provides a good checklist to review with an instrument manufacturer to fully understand the instrument's capabilities.

3.1 INSTRUMENT COMPARISONS

In order to help the reader better understand vibration measurement instrumentation, a list of common capabilities for various instrument types will be presented in this section in a glossary type format. Table I will present each type of instrument and list the general capabilities each possesses using the comparison capabilities defined below:

A) Portability - Can the equipment be easily carried around a plant or mill? How much does it weigh?

B) Typical Frequency Range - Describes in what frequency range the instrument can normally measure (usually +/-10%; or +/-3dB which equals +41%, - 30%). Outside this quoted range, the instrument will normally fall off in response, and sometimes quite sharply. This will determine if the instrument has the capability to analyze the machinery in the plant based on the frequencies

© Copyright 2001 Technical Associates Of Charlotte, P.C.

generated by sources within it. Particularly watch out for either very low-speed equipment or machines having very highspeed components. Low frequency sources might include machinery with operating speeds below 100 RPM which might have hunting tooth gear frequencies, rolling element bearing cage frequencies, subharmonics far below running speed, etc. High frequencies might include gear mesh frequencies (and their harmonics), blade pass frequencies, rotor bar passing frequencies, etc. Both the analyzer and the transducer must be capable of accurately measuring and clearly displaying each of these frequencies on the machinery to be evaluated.

C) Data Measurement Format - Describes which parameter(s) each instrument type normally can measure and display for the user as follows:

1) **Overall Level (OL)** - Only the overall vibration amplitude is reported which cannot be broken down by its frequency components.

2) **Swept Filter (SF)** - A constant percentage filter (typically 2 to 10%) is swept throughout the frequency range of the instrument (typically from about 60 to 600,000 CPM in most of these instruments).

3) **Frequency Spectrum (FS)** - Displays amplitude versus frequency for each of the peaks in the spectrum.

4) **Time Waveform (TWF)** - Displays the time (seconds or minutes) versus vibration amplitude (mils, in/sec, or g).

D) Typical Display Types - Describes the type of display used by the instrument screen. Basically, there are three different types:

1) **Liquid Crystal Display (LCD)** Good display which is usually fairly readable under good lighting conditions depending on the type of LCD display used as well as the number of horizontal and vertical pixels used in the display. Currently, supertwist displays are probably the best LCD performer.

2) **Monochrome Screen (MS)** -Typically a black or green background screen with text and figures plotted in a single color (usually white, green or orange).

3) **Analog Meter (AM)** - Vibration amplitudes printed on a fixed background with an oscillating analog needle reporting the actual amplitude being measured.

E) Typical Transducer Types - Type of vibration pickup which generally is attached to the instrument for detecting the vibration of the equipment. See Chapter 3 for a detailed explanation of these vibration pick-ups.

1) **Accelerometer (A)** - Measures acceleration (g, in/sec^2, or mm/sec^2)

2) **Velocity Transducer (V)** - Measures velocity (in/sec or mm/sec)

3) **Proximity Eddy Current Probe (P)** - Measures shaft displacement (mils or microns)

F) Phototach and/or Strobe Light Capabilities - The capability to read the input from a phototach or strobe light to determine actual 1x RPM or for use in measuring phase. Sometimes, phase might also be desired at higher operating speed multiples (2x RPM, 3x RPM, etc.)

G) Multi-Channel Availability - The capability to attach and make measurements with more than one vibration pickup at a time.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

TABLE 1. TYPICAL VIBRATION MEASUREMENT INSTRUMENT CHARACTERISTICS @ 6/97

CHARACTERISTICS	OVERALL LEVEL METERS	SWEPT-FILTER ANALYZERS	FFT DATA COLLECTORS	REAL-TIME ANALYZERS	TAPE RECORDERS
A. PORTABILITY	YES	SEMI	YES	SEMI	SEMI
Approximate Weight (Lb)	1-2	25-45	5-10	25-60	20-40
B. TYPICAL FREQUENCY RANGE (CPM)					
Low End	120	60	60	0	VARIES
High End	600,000	600,000	2,400,000	6,000,000	VARIES
C. DATA MEASUREMENT FORMAT	OL	OL, SF	OL, FS, TWF	OL, FS, TWF	-
D. TYPICAL DISPLAY TYPES	AM	AM	LCD	MS TYPICALLY	-
E. TYPICAL TRANSDUCER TYPES	PER MANUF	A,V	A, V, P	A, V, P	A, V, P
F. PHOTOTACH AND/OR STROBE LIGHT	NO	YES	MOST USE BOTH	P'TACH-YES STROBE-NO	P'TACH-YES STROBE-NO
G. MULTI-CHANNEL AVAILABILITY	NO	LIMITED	SOME	YES	YES
H. SPIKE ENERGY, HFD, OR SHOCK PULSE	SOME	SOME	MOST	NO	NO
I. HIGH FREQUENCY ENVELOPED SPECTRUM	NO	NO	MOST	FEW	NO
J. SPECTRAL DISPLAY UPDATE	-	-	LT	RT	-
K. EASE OF USE	SIMPLE	SIMPLE	FAIRLY SIMPLE	DEMANDING	VARIES
L. TIME WAVEFORM STORAGE CAPABILITY	NO	HC ONLY	MOST	YES	YES
M. FREQUENCY SPECTRA STORAGE	NO	NO	YES	YES	-
N. PREDICTIVE MAINT. SOFTWARE AVAILABILITY	NO	NO	YES	SOME	-
O. NATURAL FREQUENCY TESTING CAPABILITY	NO	LIMITED	GOOD	SUPERIOR	-
P. OPERATING DEFLECTION SHAPE (ODS)	NO	NO	YES	SOME	-
Q. EXPERIMENTAL MODAL ANALYSIS CAPABILITY	NO	NO	FEW	MOST	-
R. SYNCHRONOUS TIME AVERAGING (STA)	NO	NO	SOME	YES	-
S. WATERFALL PLOTTING CAPABILITY	NO	NO	FEW	MOST	-
T. RELATIVE COSTS					
Low End	$400	$1,000	$3,500	$4,000	$5,000
High End	$2,800	$9,000	$18,000	$40,000	$100,000
Nominal Expected Cost	$1,200	$6,000	$9,000	$15,000	$15,000

ABBREVIATIONS

A - ACCELEROMETER
AM - ANALOG METER
FS - FREQUENCY SPECTRA
HC ONLY - PRINTED HARD COPY ONLY
LCD - LIQUID CRYSTAL DISPLAY
LT - LIVE TIME (TYPICALLY UPDATED EVERY 1-4 SECONDS)
V - VELOCITY TRANSDUCER
MS - MONOCHROME DISPLAY
OL - OVERALL LEVEL
P - PROXIMITY PROBE
RT - REAL-TIME (HIGH SPEED SCREEN REFRESH RATES)
SF - SWEPT FILTER
TWF - TIME WAVEFORM

NOTE: THE ABOVE TABLE HAS NOT TAKEN INTO ACCOUNT ALL OF THE VARIOUS INSTRUMENTS AVAILABLE ON THE MARKET TODAY. DUE TO THE RAPID ADVANCES IN TECHNOLOGY, MANY PIECES OF EQUIPMENT CANNOT BE CLEARLY DEFINED INTO THE ABOVE CATEGORIES. THIS TABLE IS ONLY MEANT TO BE USED AS AN EXAMPLE OF THE CAPABILITIES A USER WOULD EXPECT TO FIND FROM A PARTICULAR INSTRUMENT TYPE. THE ABOVE CHARACTERISTICS SHOULD BE DISCUSSED IN DETAIL WITH A PARTICULAR MANUFACTURER BEFORE PURCHASING.

NOTE: "CHARACTERISTIC" IS DEFINED FOR EACH LISTED ITEM IN SECTION 3.1. DISCUSSIONS COVERING THE GENERAL CAPABILITIES AND LIMITATIONS OF EACH INSTRUMENT TYPE FOLLOW IN SECTION 3.2.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

H) Spike Energy, HFD, or Shock Pulse (SPM) Measurement Capability - Ability to provide the overall level of Spike Energy, HFD (High Frequency Detected acceleration within a specified frequency range), or Shock Pulse.

I) High Frequency Enveloped Spectrum Measurement Capability - Ability to provide an enveloped spectrum of spike energy, acceleration, etc. These are referred to by vendors as a "Spike Energy Spectrum[1]", "Amplitude Demodulated Spectrum[2]", or "Acceleration Enveloped Spectrum[3]", which normally are often measured within the 500 - 50,000 Hz frequency range (30,000 - 3,000,000 CPM). Related to these is the "SEE" spectrum developed by SKF Condition Monitoring, but this parameter is measured in the 250,000 - 350,000 Hz range (15,000,000 - 21,000,000 CPM).

J) Spectral Display Update - How fast does the screen refresh itself with up-to-date data. Sometimes the faster update speed may help in analysis problems, more clearly showing the problem and causing less frustration, particularly when performing natural frequency tests, or when examining electrical and/or beat frequency vibration problems.

Live Time (LT) - Screen updates approximately every 1-4 seconds depending on the particular model and settings of the instrument such as F_{MAX}, lines of resolution, and overlap processing percentage.

Real Time (RT) - Screen updates almost instantaneously, particularly in higher frequency spans. Real-Time Analyzers normally allow the lowest possible F_{MAX} setting that will still allow real-time screen updates (screen refresh rate still depends on setup parameters in an RTA just as it does in a data collector).

K) Ease of Use - An assessment of how long it usually requires to learn how to effectively operate the instrument; how much training is required; how meaningful the help screens are; and about how long it requires an analyst to become proficient at data collection or analysis with it. Also, an assessment of how demanding it is and if additional training is usually required if the instrument is only used every few weeks or months.

L) Time Waveform Storage Capability - The capability to store the time waveform signals received from the transducer for later analysis.

M) Frequency Spectra Storage - The capability to store the frequency spectra required for later analysis.

N) Predictive Maintenance (PMP) Software Compatible - Is there software available for trending and setting overall and spectral alarms for machinery in a predictive maintenance program as well as when setting up measurement routes?

O) Natural Frequency Testing Capability - The capability to conduct natural frequency tests with the instrument which may include "bump" or "impulse" tests, coastdown/runup tests, Bode' or Polar plot measurement (amplitude vs. frequency vs. phase), etc.

P) Operating Deflection Shape (ODS) Capability - Ability to simultaneously measure phase and amplitude at a particular frequency and "download" this information into a personal computer outfitted with the proper software to produce operating deflected shape plots using animated plots on a computer monitor. This will actually show the machine in motion at a certain frequency such as 1x RPM (or at higher running speed multiples if the user desires and if his instrument can do so at these higher frequencies).

Q) Experimental Modal Analysis Capability - Describes the typical capability of each instrument type to measure natural frequencies, mode shapes and transfer functions. This

© Copyright 2001 Technical Associates Of Charlotte, P.C.

can often lead to a rather quick, inexpensive solution such as advising the user to weld a brace with just the right geometry and mass to resolve a natural frequency problem. Modal Analysis requires at least a two channel analyzer and modal software.

R) Synchronous Time Averaging (STA) Capability - The capability to perform Synchronous Time Averaging with the end result being that all frequencies which are not exact multiples of a designated frequency are eliminated. These nonsynchronuous frequencies include running speed multiples of shafts operating at a different RPM, 2x Line Frequency, bearing defect frequencies, belt defect frequency or speed, etc. Note that these nonsynchronuous frequencies will disappear from both the spectrum and time waveform if a sufficient number of averages are taken (often 250 to 500 averages). This can be of great assistance to the analyst attempting to evaluate a machine with significant vibration transmitted into it from its neighboring machines from a felt on a paper machine, or even from nonsynchronous frequency components within the machine rotor itself being evaluated since again it will limit the vibration spectrum being measured only to multiples of the frequency which are synchronous with the trigger source (such as a phototach or strobe light).

S) Waterfall or Cascade Plotting Capability - The capability to display either data collected during a “run- up” or “coast-down”; or data collected from one PMP survey to the next directly on the spectral display of the instrument in question.

T) Relative Costs - An estimate of the current costs for these types of machines. These ranges may vary considerably, based on the “extras” such as software, cabling, number of channels, and auxiliary equipment which may be purchased. Please note the “Nominal Cost” which is what one should expect to pay for most of the instruments of each type, and outfitted with the usual number of “bells and whistles” for an instrument in this class.

3.2 GENERAL CAPABILITIES OF EACH VIBRATION INSTRUMENT TYPE

The following is a summary of the instruments covered previously in Table I outlining both their major advantages and drawbacks.

3.21 OVERALL LEVEL VIBRATION METERS

Overall level vibration meters are used to measure so-called total vibration in the form of acceleration, velocity, or displacement which is then most often displayed on an analog meter (many of these meters also measure one of the high frequency banded parameters such as spike energy (gSE)). They are lightweight and very portable which makes them ideal for collecting data from in-service machinery around a plant or mill. These instruments have been around for years and, at one time, were the main instruments used to evaluate large quantities of machines in typical predictive maintenance programs of yesteryear (the analyst recorded his measurements on a clipboard taken on the route). Plots of overall vibration versus time (trend plots) were then manually generated using these results.

However, the overall level vibration meter is not now a very useful tool in a vibration monitoring/analysis program due to its lack of capability. They are only possibly useful in monitoring non-critical machinery since the vibration level at specific frequencies cannot be measured by this instrument. In this case, potentially severe problems can develop which have negligible effect on the overall level (i.e., bearing wear, gear wear, cracked gear teeth, cracked rotor bars, etc.)

The overall vibration meter does not have the capability to measure vibration amplitude versus frequency content (frequency spectrum).

© Copyright 2001 Technical Associates Of Charlotte, P.C.

Frequency content is very important since even very low levels of vibration at certain frequencies can signify definite machine problems. Most of these instruments likewise do not have the capability of storing the measured vibration signal or frequency output. The vibration reading must be manually recorded creating a cumbersome data collection procedure. Also, the frequency range of the instrument may be limited and should be checked before use. Furthermore, it is not capable of displaying the time-based waveform (amplitude vs. time). Worst of all, numerous failures can occur with only a minute increase (or decrease) in the overall level, particularly dependent on which vibration parameter is being measured and trended (displacement, velocity, or acceleration).

Finally, even in the case of overall level meters which can measure the high frequency banded acceleration (such as overall spike energy), these instruments can even be rendered somewhat ineffective. For example, if the user feels he can monitor bearing health simply by monitoring overall spike energy, HFD, or shock pulse, he should be aware that each of these are effected not only by rolling element bearing condition, but also by lubrication, cavitation, high pressure steam or air, gear condition, rotor rub and belt squeal. To determine what is causing this parameter to increase will require spectrum analysis (hopefully using this parameter vs. frequency).

3.22 SWEPT-FILTER ANALYZERS

Swept filter analyzers are normally an older technology instrument which are an enhancement over the overall level vibration meter alone by providing a means for extracting the filtered vibration level at various frequencies. It is used by adjusting a tuning knob which filters out any vibration signals outside the filter frequency. For example, using a 10% swept filter, measuring a frequency at about 1000 CPM would actually include the amplitude within the 950 to 1050 CPM range. Only the vibration level for the frequency range being swept is reported on its analog meter display. Typically, the filters employed on these units vary from approximately 2% to 10%, meaning that frequency precision is limited (and, the higher the frequency, the lower the precision of the frequency reading in terms of CPM or Hz).

This type of equipment has nearly been replaced by the capabilities of the newer FFT data collectors with higher precision frequency spectrum and also data storage capabilities. However, they can still be of some use for field balancing, strobe light “slow motion studies”, phase analysis with the use of the strobe, determining natural frequencies of components (to some extent) and determining vibration amplitude at various frequencies. Outputs to an analog X-Y plotter are available on some of these units, and even built in on others.

These instruments are typically large and heavy in size, and not very portable. This is acceptable for analyzing one machine. However, it now would be a nightmare to use them exclusively on a predictive maintenance route. The data cannot be stored for further analysis or off-loaded to a computer. The entire process of collecting frequency data is very time consuming and cumbersome. Also, they are not normally capable of displaying the time-based waveform.

3.23 FFT PROGRAMMABLE DATA COLLECTORS

FFT data collectors are the current state-of-the-art instruments of choice for predictive maintenance programs and also can be useful in a variety of analysis situations. The FFT (Fast-Fourier Transform) capability allows the time waveform captured by these units to be transformed into a frequency spectrum and displayed on a small LCD screen in live-time (typically within 1-4 seconds depending on the data collection mode, along with the chosen frequency span, and lines of resolution settings on the instrument). The control panel allows rapid switching of the display parameters which is helpful when performing monitoring on many different machines and their components, or when performing analysis on troubled

© Copyright 2001 Technical Associates Of Charlotte, P.C.

equipment. Also, increasing the speed of some data collectors is the ability to simultaneously measure 3 different parameters with one push of the "store" button (such as velocity, acceleration, and spike energy).

The data collector was designed for data collection on many pieces of machinery within one outing. It is portable and has the capability of storing data for further analysis back at the office. Typically, software designed to work with the data collector is installed on a personal computer which can download a route into or receive data from the data collector for analysis of the machine's condition. Also, the data collector can be used as an analyzer in the field due to its graphics, FFT, and live-time capabilities. Most of them are also capable of time waveform presentation. Furthermore, a phototach or strobe light can be attached to most of today's data collectors to measure phase at various points. Most of them can also measure high frequency parameters such as HFD, Spike Energy, etc., used for identification of problems such as bearing deterioration, worn gear teeth, lubrication problems, cavitation, and rubs. In addition, many of these units can now generate a spectrum of one of these high frequency parameters (i.e., spike energy spectrum①, amplitude demodulated spectrum②, acceleration enveloped spectrum③, etc.)

The LCD display of some data collectors is "live-time" which typically updates on the average of once every 1 to 4 seconds as compared with a real time spectrum analyzer which has nearly instantaneous updates (again depending on frequency range). Note that some data collectors have markedly faster live-time update speeds than others. Only a few FFT data collectors are capable of multi-channel data input which is helpful in operating deflection shape tests or modal analysis, as well as for greater data collection speed.

The frequency range of the average data collector is normally limited to about 60 CPM (1 Hz) on the low end and up to about 1,500,000 CPM (25,000 Hz) on the high end which may pose some analysis or data collection problems. However, some data collectors now are available which allow measurements down to as low as approximately 6 CPM (0.10 Hz) and up to as high as 4,500,000 CPM (75,000 Hz). Real-time spectrum analyzers can sometimes measure frequencies down to almost 0 CPM, and up to about to 6,000,000 CPM (0 - 100,000 Hz). With the continued intense development of data collectors taking place today, no telling what their capabilities might be within the next 5 to 10 years!

In terms of memory, one of the more major enhancements to some of the data collectors has been the addition of PCMCIA cards, which give them practically unlimited storage. Also, some of these PCMCIA cards allow for the loading of special functions into the instruments such as balancing, transient capture, natural frequency testing, etc.

Notes:

①= Spike Energy Spectrum (Available from IRD Mechanalysis, based in Columbus, OH)

②= Amplitude Demodulated Spectrum (Available from CSI, based in Knoxville, TN)

③= Acceleration Enveloped Spectrum (Available from SKF Condition Monitoring based in San Diego, CA)

3.24 REAL-TIME SPECTRUM ANALYZERS

The real-time spectrum analyzer is the most powerful diagnostic tool on the market for advanced diagnostic techniques. There are a variety of them now available having a wide range of capabilities. In layman's terms what is meant by a "real-time" analyzer is that the user is usually able to simultaneously see the oscillations of all peaks within the frequency span chosen. That is, if one had a 2-pole, 3590 RPM motor belt driving a 1000 RPM fan in close proximity to another fan operating at about 1050 RPM, the user should be able to see two sets of pulsating

© Copyright 2001 Technical Associates Of Charlotte, P.C.

frequencies - one set would be the two fan speeds of 1000 and 1050 RPM which would likely set up a “beat frequency” of 50 CPM thereby causing the amplitudes of both the 1000 and the 1050 RPM peaks to pulsate up and down; the second set of pulsating frequencies would likely be located at 2x RPM of the motor (2 x 3590 = 7180 CPM) and the frequency at 2x electrical line frequency (2 x 3600 = 7200 CPM). If one set up the proper frequency range and number of FFT lines, he should readily see this other peak beating between 7180 and 7200 CPM frequencies.

The typical real-time analyzer is capable of measuring frequencies in a wide range from about 0 to 6,000,000 CPM (0 - 100,000 Hz) which is important for diagnosing gear problems and blade pass frequency problems on high-speed machines (however, the limitations here will most always be the frequency range capabilities of the transducer employed, as well as the method used to mount the transducer itself). Multi-channel capabilities, available in many of these units, allow excellent data capture for operating deflection shape and modal analysis, as well as speeding up the process of data collection. Most all real-time analyzers are also typically capable of performing synchronous time averaging (STA), order tracking, and are able to perform a wide variety of natural frequency tests.

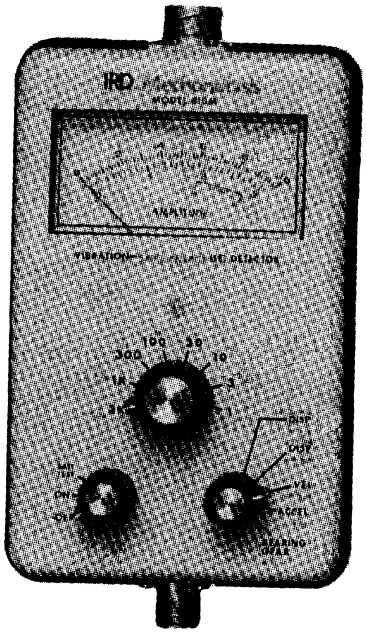
The real-time display can be very helpful in diagnosing problems as opposed to the live-time display in data collectors. The screen updates “faster than the eye” in a real-time analyzer when the F_{MAX} is properly specified. Photo-tach input is available for measuring and comparing phase between measured points and for many different frequencies simultaneously. Real-time analyzers are excellent instruments for performing impulse natural frequency tests due to their “peak hold” capabilities. Furthermore, “peak hold” can be used to effectively monitor coast-down and run-up tests when attempting to identify system natural frequencies. Also, Bode', Polar and Waterfall plots can be generated using real-time analyzers as another verification of natural frequencies (as well as identifying the component(s) which is resonant). These analyzers can likewise be used in a fashion similar to oscilloscopes to generate shaft orbit plots, computerized order tracked spectra, etc. Some of these analyzers are also outfitted with digital time buffers which can be used in a manner similar to tape recorders, capturing transient events, runups, coastdowns, etc. from data captured and stored to hard disk in the field (giving such data filenames so they can be evaluated and reported upon at covenience by the analyst).

The real-time spectrum analyzer is a more complex piece of equipment which requires additional training and frequent use to remain proficient in its use. It is not very portable and should only be used for machinery trouble-shooting and specialty monitoring. Memory is typically limited in comparison with the PMP data collectors. However, some of them are now outfitted with 3.5 inch (1.44 Mb) floppy drives which gives them almost unlimited spectral and time waveform storage. Also, some real-time analyzers are now composed of a computer out-fitted with special cards, making them an even more useful instrument and providing them with memory limited only by hard disk size. In these cases, it is sometimes possible to install PMP software on some of these units thereby allowing their spectra to be passed back into the PMP database. Finally, some of these computerized real-time analyzers allow use of word processing, spread-sheets and graphics software thereby allowing “on-the-spot” report generation if desired.

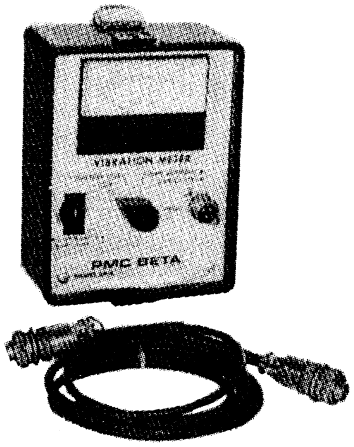
3.25 INSTRUMENT QUALITY TYPE RECORDERS

Instrumentation quality tape recorders are useful devices for machinery troubleshooting and for simultaneously recording many different signals (whether from vibration pickups, or from pressure transducers, tachometer signals, current transformers, phototachs, etc.). Both analog and digital tape recorders are available. The analog type records the actual signal input from the transducer without breaking it into a number of sampled points similar to a voice recorder. However, analog units today are usually limited to a dynamic range of only about 40 to 48 dB. The digital recorder samples the input at a specified rate and reproduces it as stored numbers with the advantage of a much larger dynamic range capability (now on the order of about 78 dB). That

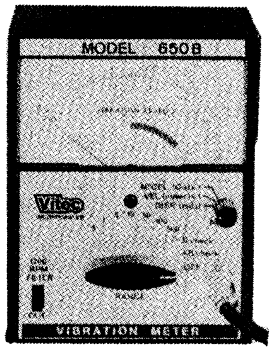
© Copyright 2001 Technical Associates Of Charlotte, P.C.



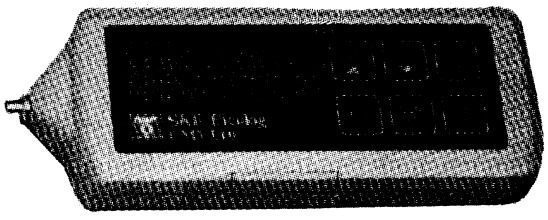
A. IRD 810 VIBRATION SPIKE ENERGY DETECTOR



B. PMC/BETA 201A VIBRATION METER



C. VITEC 650 VIBRATION METER

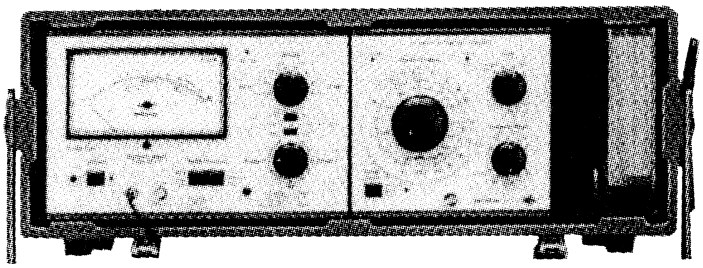


D. SKF CMVLIO PICOLOG VIBRATION / SEE / ACCELERATION ENVELOPE DETECTOR

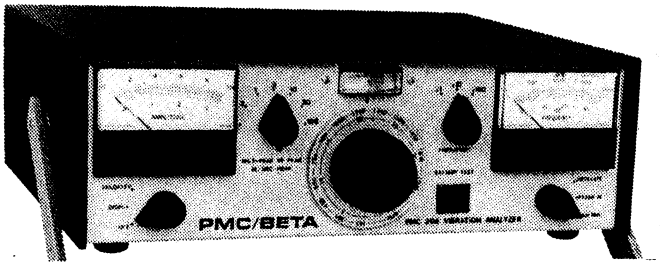
E. SKF CMVP20 VIBRATION PEN

FIGURE 1. EXAMPLES OF VARIOUS HAND-HELD OVERALL VIBRATION METERS

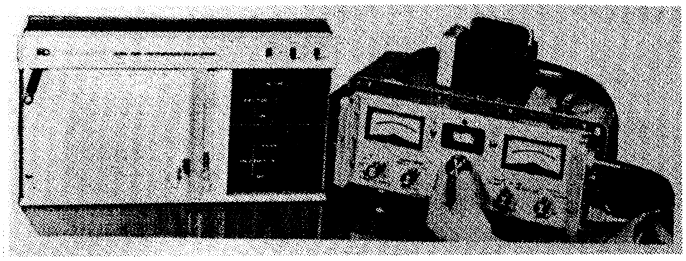
© Copyright 1997 Technical Associates Of Charlotte, P.C.



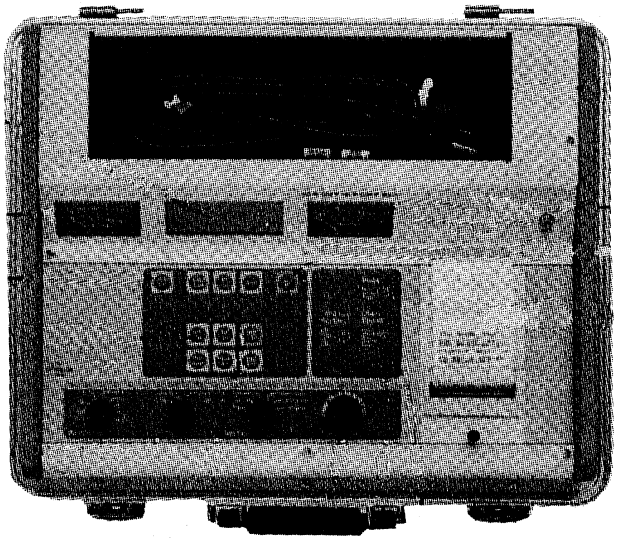
A. B&K 3513



B. PMC/BETA 208



C. IRD 350 WITH
IRD 1080 PLOTTER



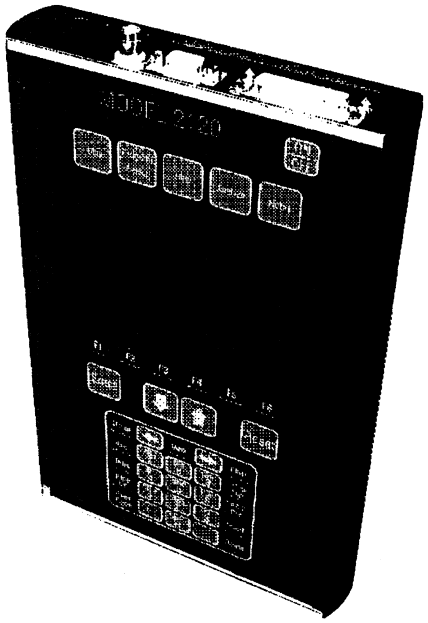
D. IRD 880

E. SCHENCK VIBRATION 30

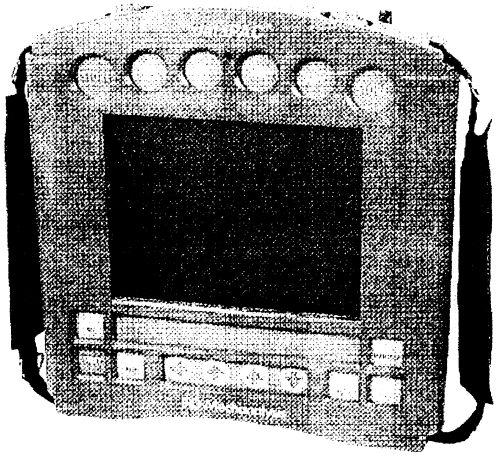
F. VITEC VIBRA/VIEW

FIGURE 2. EXAMPLES OF PORTABLE SWEPT-FILTER ANALYZERS

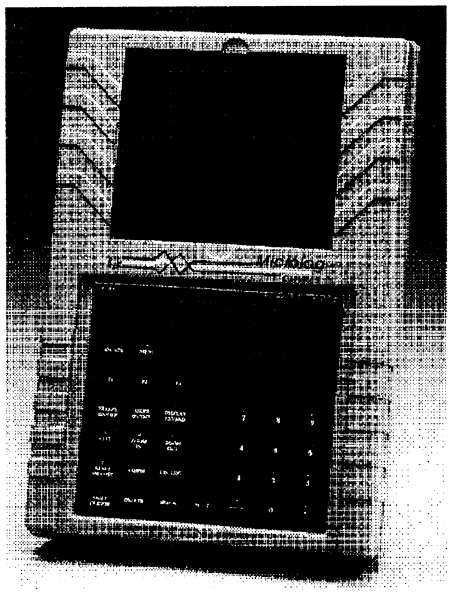
© Copyright 1997 Technical Associates Of Charlotte, P.C.



A. CSI 2120
MACHINERY ANALYZER



ENTEK IRD DataPac 1500

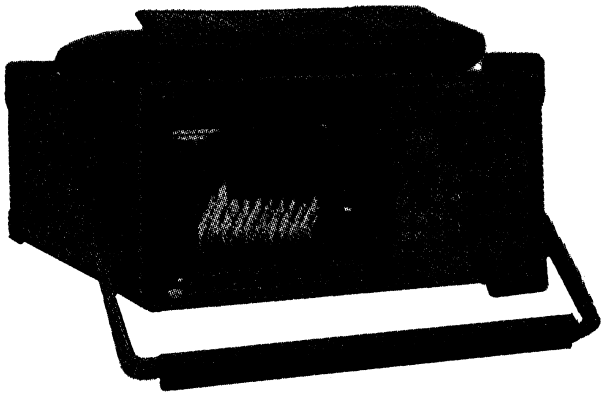


C. SKF MICROLOG CMVA55

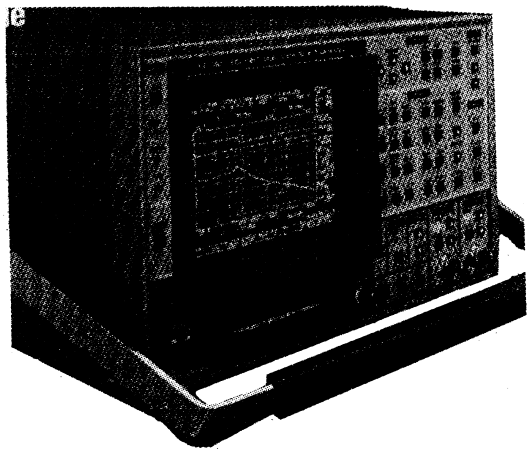
D. PREDICT/DLI WATCHMAN DC-7B
(3 CHANNEL)

FIGURE 3. EXAMPLES OF VARIOUS DATA COLLECTORS

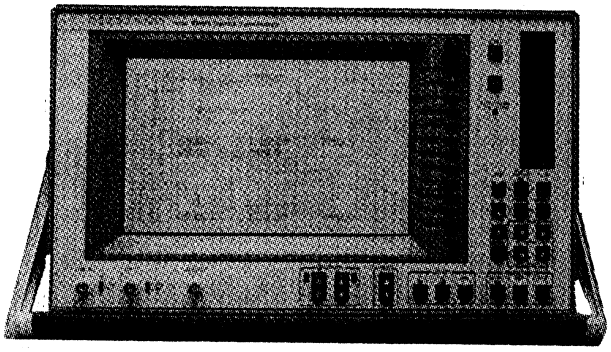
© Copyright 1997 Technical Associates Of Charlotte, P.C.



A. HEWLETT PACKARD 35670A
DYNAMIC SIGNAL ANALYZER
(2 OR 4 CHANNEL)



B. ONO SOKKI CF-360
(2 CHANNEL)



C. ROCKLAND 90665L
DYNAMIC SIGNAL ANALYZER
(2 CHANNEL)

D. SIGNAL PROCESSING SYSTEMS SA 390
DYNAMIC SIGNAL ANALYZER
(2, 4, 6, OR 8 CHANNEL)

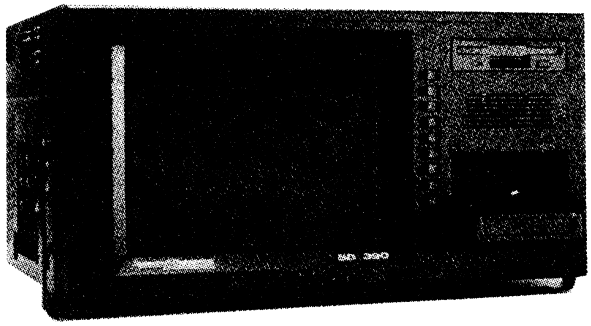
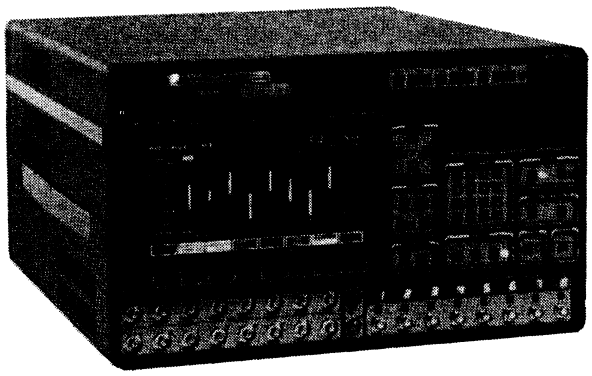
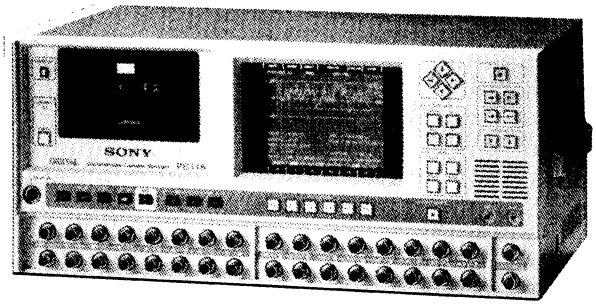


FIGURE 4. EXAMPLES OF VARIOUS REAL-TIME SPECTRUM ANALYZERS

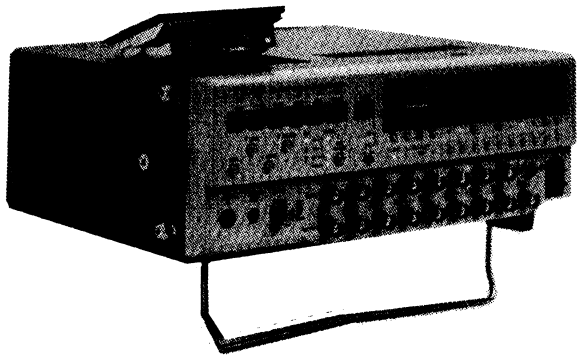
© Copyright 1997 Technical Associates Of Charlotte, P.C.



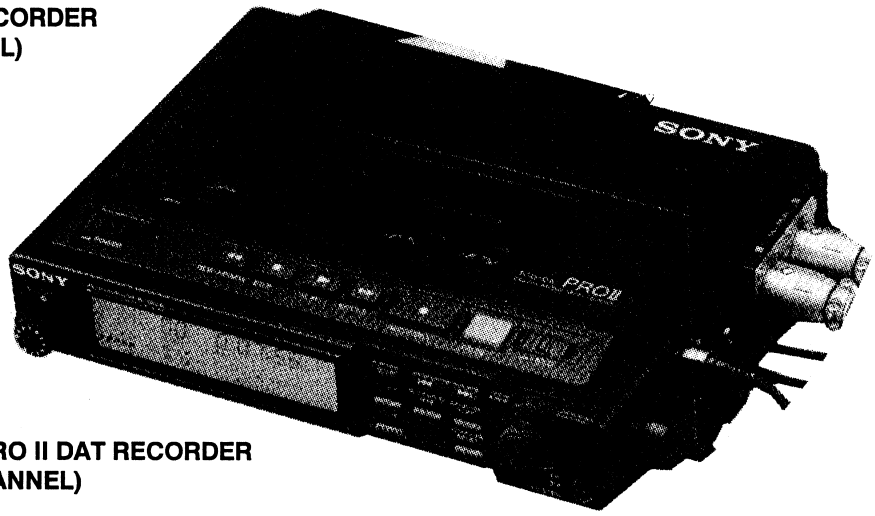
A. METRUM RSR 512 DAT RECORDER
(8,16,24 OR 32 CHANNEL)



C. SONY PC 116 DAT RECORDER
(2 THRU 16 CHANNEL)



B. TEAC RD-130T DAT RECORDER
(2, 4, OR 8 CHANNEL)



D. SONY TCD-D10 PRO II DAT RECORDER
(2 CHANNEL)

FIGURE 5. EXAMPLES OF INSTRUMENT QUALITY TYPE RECORDERS

© Copyright 1997 Technical Associates Of Charlotte, P.C.

is, digital tape recorders commonly have more than 30 dB higher dynamic range, which means they are more than 31 times as amplitude sensitive (and therefore can detect low amplitude frequencies such as bearing frequencies of only about 0.01 in/sec in the presence of much higher amplitude peaks such as a running speed amplitude of more than 1.0 in/sec). Such a capability would be out of the question with the older analog tape recorder models.

Tape recorders allow the analyst the capability of very accurately recording the time waveform of the vibration signal for future analysis with measurements made by a real time spectrum analyzer. In addition, they afford him the luxury of employing a wide range of frequency range spectra back at the office. Both types of tape recorders have multi-channel capabilities (up to 64 channels or more) which allow the user to capture many data points simultaneously. This is especially useful when many points need to be analyzed, or if machine testing requires that the machine be stopped resulting in lost production. Input can be brought in from a wide variety of signals such as voice or noise, pressure, tachometer signal (RPM), vibration, or electrical current from which FFT spectra can be generated and evaluated. Tape recorders will also capture short-lived transient events which cannot even be “seen” by an analyzer. After the signal has been captured on tape, the user has excellent flexibility in analyzing the signal back in the office since it may be played back over and over again looking at alternate spectra settings. For instance, frequency span can be adjusted to provide the necessary resolution. Also, anomalous events can be analyzed in many different settings to determine the source.

However, depending on the capabilities of the recorder (analog vs. digital, dynamic range, etc...), instrument quality tape recorders can be very costly. Furthermore, they may be complicated to use and could require frequent use to remain proficient in data collection.

REFERENCES

1. “Spike Energy Spectrum” - developed by IRD Mechanalysis, Inc., based in Columbus, OH.

2. “Amplitude Demodulated Spectrum” - developed by Computational Systems, Inc. (CSI), based in Knoxville, TN.

3. “Acceleration Enveloped Spectrum" - developed by SKF Condition Monitoring with USA headquarters in San Diego, CA.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

CHAPTER 4

OVERVIEW OF VARIOUS VIBRATION TRANSDUCERS AND HOW TO PROPERLY SELECT THEM

4.0 INTRODUCTION

The purpose of **Chapter 4** is to introduce the reader to vibration transducers, the various types available, how they function, how to select them and to provide information on transducers currently available on the market today. Each of the following transducers will be covered:

1) **Accelerometers**
2) **Velocity Pickups**
3) **Non-contact Eddy Current Displacement Probes**
4) **Shaft Contact Displacement Probes (including Shaft Sticks and Shaft Riders)**

Figure 1 illustrates the three most common vibration transducers in use today, including the non-contact displacement probes, velocity pickups and accelerometers. Also shown are the components making up these transducers, each of which will be discussed in more detail under separate sections. Table I includes a general summary of some of the more important characteristics and general specifications for each of the various categories of accelerometers, velocity pickups and non-contact probes. In addition, it includes example models of transducers made by various manufacturers.

Depending on the measurement to be made, each of these transducers have their own optimum applications which will be discussed in this chapter. Also, particularly in the case of accelerometers, there is a wide variety of individual transducers. Finally, mounting techniques and their influence on the vibration measurement accuracy will likewise be discussed in this chapter.

As with **CHAPTER 3 - "OVERVIEW OF THE STRENGTH AND WEAKNESSES OF TYPICAL VIBRATION INSTRUMENTS"**, it is important to note that the following information is meant to reflect the general state of transducer design. However, the specific design characteristics for each individual transducer may vary from vendor to vendor. The reader should use this chapter as a guide for discussing specific needs with a vendor before purchasing a particular transducer or set of transducers.

4.1 TYPES OF VIBRATION TRANSDUCERS AND THEIR OPTIMUM APPLICATIONS

Typically, there are four types of vibration measurement transducers. They are known as accelerometers, velocity pickups, non-contact eddy current displacement probes and shaft contact displacement probes (including shaft sticks and shaft riders). Each type of transducer has applications which justify its use for monitoring machinery. These applications are important to understand so that a vibration analyst may best take advantage of them. For instance, when ordering large, high-speed machinery (i.e. turbine-generators or centrifugal air compressors), a

© Copyright 2001 Technical Associates of Charlotte, P.C.

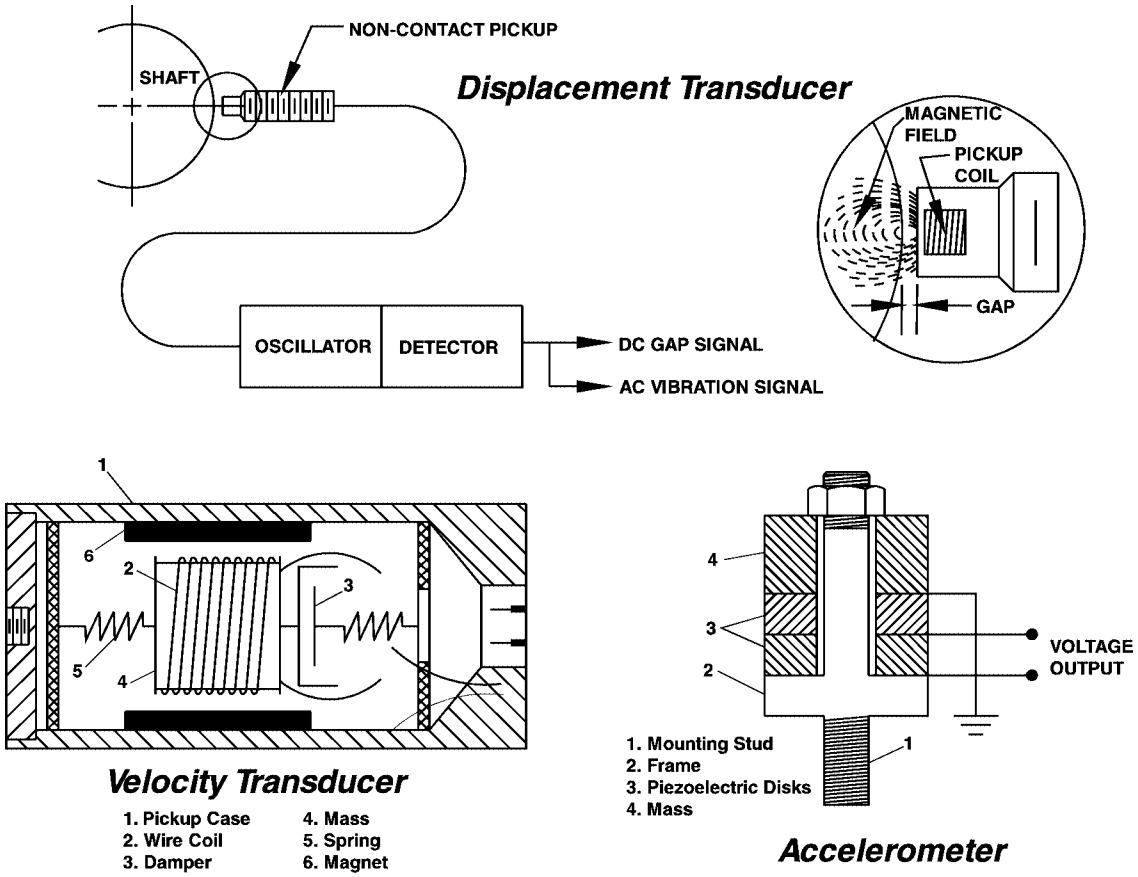


FIGURE 1
EXAMPLES OF VARIOUS TYPES OF VIBRATION TRANSDUCERS (REF. 5)

© Copyright 2001 Technical Associates of Charlotte, P.C.

© Copyright 2001 Technical Associates of Charlotte, P.C.

TABLE I
GENERAL TRANSDUCER CHARACTERISTICS

TYPICAL SPECIFICATIONS	ACCELEROMETERS					VELOCITY PICKUPS		NONCONTACT DISPLACEMENT PROBES
	GENERAL PURPOSE	LOW FREQUENCY	HIGH FREQUENCY	PERMANENTLY MOUNTED	TRIAXIAL	SEISMIC	PIEZOELECTRIC	
TYPICAL SENSITIVITY RANGE (mV/g Unless Noted)	10 – 100	500 – 10,000	0.4 – 20	10 – 100	10 – 100	(mV/in/s) 100 – 1000	(mV/in/s) 100 – 1000	(mV/mil) 100 – 200
TYPICAL MEASUREMENT FREQUENCY RANGE (CPM)	60 – 600,000	6 – 60,000	600 – 3,600,000	180 – 600,000	120 – 600,000	600 – 60,000	60 – 270,000	0 – 120,000
TYPICAL STUD MOUNTED NATURAL FREQUENCIES (CPM)	960,000 – 2,700,000	390,000 – 2,100,000	4,200,000 – 10,800,000	1,080,000 – 1,800,000	480,000 – 1,200,000	APPROXIMATELY 200,000	APPROXIMATELY 200,000	N.A.
TYPICAL WEIGHT RANGE (grams)	17 – 110	135 – 1000	1.2 – 15	60 – 180	17 – 130	100 – 150	150 – 250	230 – 280
TYPICAL USABLE TEMPERATURE RANGE (°F)	(–100°) – 250°	(–100°) – 250°	(–40°) – 350°	(–50°) – 250°	(–100°) – 250°	(–50°) – 250°	(–50°) – 250°	(–30°) – 350°
EXAMPLE TRANSDUCER MODEL NUMBERS AND MANUFACTURER (Alphabetical Order)	B & K 4370/81 B & K 4390 IRD 943 IRD 970 PCB 302A06 PCB 302A09 PCB 307A PCB 308B RION PV–44A VIBRAMET. 1020 VIBRAMET. 1021 WILCOXON 793	B & K 8318 (S) PCB 326A01 (S) PCB 326A02 (S) PCB 326A03 (S) PCB 393C (C) VIBRAMET. 5100 (S) WILCOXON 731 (C) WILCOXON 793L (C) WILCOXON 797L (S) (S)–SHEAR MODE (NOT TEMPERATURE SENSITIVE; BRIEF SETTLING TIME) (C)–COMPRESSION MODE (TEMPERATURE SENSITIVE; LONG SETTLING TIME)	B & K 8309 PCB 303A02 RION PV 90B VIBRAMET. 6032 WILCOXON 732	B & K 8310/24 B & K 8317 IRD 941B IRD 942 PCB 328B VIBRAMET. 1024	B & K 4321 PCB 306A PCB 306A06 VIBRAMET. 3130 WILCOXON 733	BENTLY. 47633 IRD 544	BENTLY. 26862 IRD 560 WILCOXON 756V WILCOXON 793V WILCOXON 797V	BENTLY. 3300 BENTLY. 7200 IRD 403

NOTES:

1. For detailed information on particular transducers, see the APPENDIX which includes specifications for many of them.
2. By inclusion of specific transducers within this table, Technical Associates does not necessarily recommend them.

plant may wish to include internally mounted non-contact displacement probes in the specifications to measure relative shaft vibration, since much of the energy will not transmit through the heavy casings to the exterior housing where an accelerometer or velocity pickup may be magnetically mounted by PMP personnel on a data collection route. The following sections describe the most common applications for the various transducer types in use today.

4.11 ACCELEROMETERS

As the name implies, accelerometers are sensors which provide the direct measurement of acceleration (g). Typically, there are three primary performance characteristics which accelerometers possess affecting their performance. These include voltage sensitivity (mV/g), frequency response (Hz or CPM) and weight (oz. or grams). This section will take a brief look at each of these characteristics and will make some introductory remarks concerning their applications.

Figures 2, 3 and 4 illustrate the three most common types of piezoelectric accelerometers in use today (in simple terms, the piezoelectric effect of a crystal is its ability to accumulate electrical charges on its surface when the crystal is mechanically stressed). Figures 2 and 3 show one of the two major piezoelectric accelerometer types called the “compression mode accelerometer” (note that Figure 3 is an inverted compression type accelerometer). Figure 4 shows the other major type known as a shear mode accelerometer (differences of which will later be discussed). Each of the three accelerometers pictured in Figures 2 thru 4 have built-in electronics known as ICP circuits which are most often supplied with 4 to 20 milliamp (mA) current either from an external power supply or directly from the vibration instrument itself. The term “ICP” stands for “integrated circuit piezoelectric” type.

Typically, accelerometers are used with rolling element bearing supported shafts. This is because rolling element bearings transfer the majority of the shaft vibration into the machine housing and can be readily picked up by a transducer mounted on the external housing. However, they sometimes also work well with journal bearings because of advancements made in the sensitivity of certain accelerometers allowing them to pick up low amplitude vibration. However, be aware that readings from a non-contact eddy current displacement probe (measuring actual shaft vibration) may be quite a bit higher in amplitude than that taken on the casing itself (depending on the oil film characteristics as well as on the mass and distance from the shaft out to the bearing housings). Sometimes, the shaft vibration may be as much as 20 times higher on a journal bearing machine at a distance of 3 to 6 feet to the bearing surface meaning that potentially serious problems such as oil whirl or oil whip may be missed altogether if readings are not taken directly from the shaft itself due to the loss of signal through the oil film, as well as at each of the metal interfaces and thicknesses out to the machine housing.

Accelerometers are the most common PMP transducers in use today due to a number of reasons. They have relatively inexpensive costs when compared to velocity pickups and have much wider frequency range capabilities than either velocity pickups, non-contact probes, shaft sticks or shaft riders. Also they can be easily mounted onto machinery by stud, adhesive, magnet or (if absolutely necessary) extension probe. On the other hand, non-contact probes must be permanently mounted to give good, repeatable data required in predictive maintenance programs (and great effort has to be spent to prepare the target area at which the probe will be pointed, ensuring a very smooth surface which is free of cracks and other defects).

There are a large variety of accelerometers available which have a wide selection of frequency ranges. This is shown in Table I which will summarize much of the information within Chapter 4, not only providing general information about all the major transducers discussed, but also specific manufacturer and model numbers of each of the various types of accelerometers, velocity pickups and non-contact probes. Table I shows that the “General Purpose Accelerometer” has a relatively wide “Typical Frequency Range” of approximately 120 to 600,000 CPM (2 - 10,000 Hz).

© Copyright 2001 Technical Associates of Charlotte, P.C.

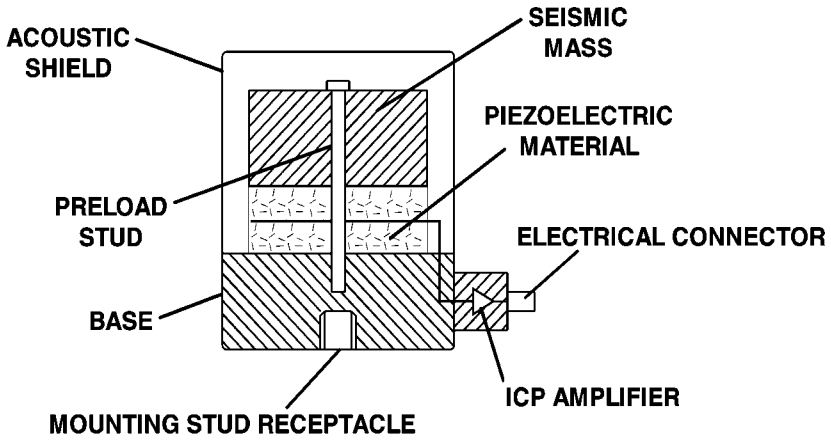


FIGURE 2
COMPRESSION TYPE ACCELEROMETER (REF. 2)

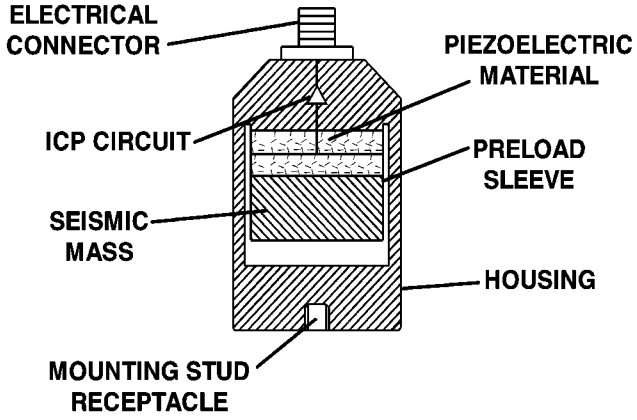


FIGURE 3
INVERTED COMPRESSION TYPE ACCELEROMETER (REF. 2)

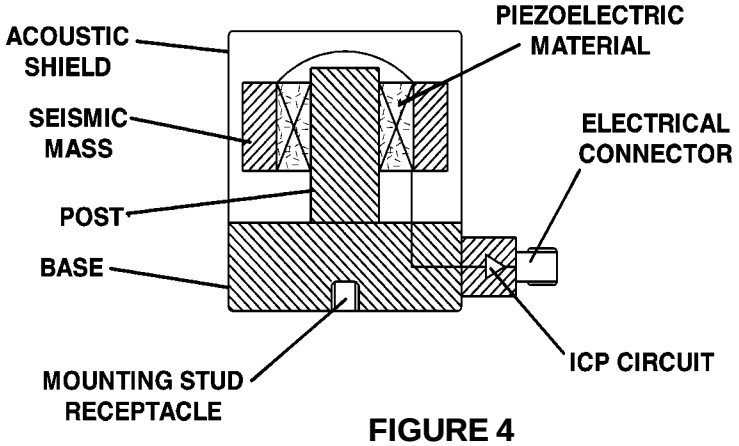


FIGURE 4
SHEAR TYPE ACCELEROMETER (REF. 2)

© Copyright 2001 Technical Associates of Charlotte, P.C.

However, it also shows that in order to make very low frequency or high frequency measurements, it will likely be necessary to acquire other specially designed accelerometers. For example, the typical low frequency accelerometer might measure down to about 6 CPM, but up to only about 60,000 CPM (0.1 - 1,000 Hz); whereas a typical high frequency accelerometer might effectively measure from approximately 600 to as high as 3,600,000 CPM (10 - 60,000 Hz).

In most cases, if desiring to take low frequency measurements, it will be necessary to choose one of the low frequency accelerometers listed in Table I which typically are much higher weight and normally have voltage sensitivities (mV/g) much higher than the general purpose accelerometers as shown in the table. The reason for this is that even large vibration displacement (mils) produces very small acceleration levels (g) when at frequencies of less than 60 CPM (1 Hz), making it of great importance to have high voltage sensitivity to bring the vibration signal above the noise. For example, an excessive displacement of 100 mils (Pk-Pk) (.100 inch) at a frequency of only 60 CPM (1 Hz) corresponds to an acceleration of only .005 g Pk at this frequency. On the other hand, if desiring to make measurements on machines generating very high frequencies (typically above 600,000 CPM or 10,000 Hz), it will likely be necessary to acquire one of the high frequency accelerometers listed in Table I. Typically, these high frequency accelerometers are much smaller accelerometers which likewise have smaller voltage sensitivity, usually on the order of only about 10 mV/g.

One of the common misconceptions is that the higher the voltage sensitivity (mV/g), the better the accelerometer. This is not the case in all measurement situations. For example, one of the problems with some of the seismic accelerometers used in low frequency measurements is that they are very sensitive to temperature changes; and, if dropped, they can fail due to the tremendous voltage response which oversaturates their built-in electronics (a general purpose or high frequency accelerometer can generally be dropped with no damage to the transducer).

Note that Table I also lists some accelerometers specially designed for permanently mounting as well as those designed to make triaxial measurements (simultaneous measurements in the horizontal, vertical and axial directions). The permanently mounted transducers are designed so that they can withstand harsh environments and sometimes can be placed under water or within other fluids such as lubricating oil or cutting oils. On the other hand, triaxial accelerometers give the analyst the advantage of simultaneously making measurements in all three directions, thereby increasing his measurement speed on a PMP route in the field.

Returning to Figures 2 thru 4, note that these illustrate the three most commonly used accelerometer designs today. Note that there are two types of compression designs (upright and inverted positioning of the seismic mass) in which the piezoelectric crystal is sandwiched between a seismic mass and a base or housing. In this case, the amount of electrical output generated by the crystal is proportional to the amount of acceleration experienced by the seismic mass which is proportional to the amount of force applied to it (as per Newton's law of motion, $F = ma$). The compression designs in Figures 2 and 3 are probably the most widely utilized designs today, mainly due to their simplicity and lower cost, along with their wide frequency range. However, one of the disadvantages with compression mode accelerometers is that they are often subject to problems due to thermal transients and base strain sensitivity which can oversaturate their electronics and require much time for the transducer to "settle".

In those situations where there are large differences in temperature on which the accelerometer will be placed (or if either the temperature of the surface on which the accelerometer will be placed is continually changing or if high pressure air is blowing on the accelerometer), the shear type accelerometer pictured in Figure 4 can help overcome these settling problems. These shear mode designs tend to isolate the crystal element from the base and housing by sandwiching the element between the seismic mass and a center post. When accelerated, the seismic mass will apply a shear force to the crystal element which will emit a voltage proportional to the acceleration (and therefore to the force) applied to it. The advantages of shear mode

© Copyright 2001 Technical Associates of Charlotte, P.C.

accelerometers include a stable output signal in the presence of thermal transients and/or high pressure air blow-by, especially when measuring at low frequencies. Also these are typically smaller in size and mass. Disadvantages of these units typically include a higher cost due to the added components required to make up the shear configuration.

Later on, when an analyst begins taking low frequency measurements (typically below 60 CPM or 1 Hz), this difference between shear mode and compression mode accelerometers will become important. In this case, it may require a compression mode accelerometer 3 or 4 minutes for it to stabilize before any measurements can be taken whatsoever. However, in these cases, a shear mode accelerometer will not be so sensitive to such thermal transients, and will stabilize quickly, allowing the analyst to begin the measurement almost immediately.

One other important difference concerns the type of signal conditioning electronics serving the piezoelectric accelerometers. There are two types of such signal conditioning including (1) high impedance, charge mode or charge type which requires an external signal conditioner; and (2) low impedance, voltage mode, integrated circuit or "ICP" (integrated circuit piezoelectric) types which contain built-in signal conditioning electronics. Each of these is pictured in Figure 5. One of the biggest disadvantages of the charge mode accelerometer is that they require a separate charge amplifier between the analyzer and the sensor as shown in Figure 5. On the other hand, ICP current supply is now included within most all FFT analyzers (including both data collectors and spectrum analyzers) allowing an analyzer to directly power the ICP, voltage-mode accelerometer. Another disadvantage with the charge mode units is that special, low-noise cable of fixed length most often has to be used between the charge amplifier and sensor in the field whereas the ICP voltage-mode sensor requires only ordinary 2-wire coaxial cable and the ICP accelerometer can usually be powered at distances up to about 1000 feet without additional power sources. In the case of the charge mode unit, accelerometer sensitivity is usually defined in units of picocoulombs per g (pC/g), whereas the sensitivity of voltage-mode units is normally expressed directly in millivolts per g (mV/g). Advantages of the charge mode accelerometer include the ability to operate at high temperatures. However, its disadvantages of requiring the low-noise, fixed length cable, along with a clean and moisture free environment as well as a separate charge amplifier do not normally lend it to most predictive maintenance measurement systems today.

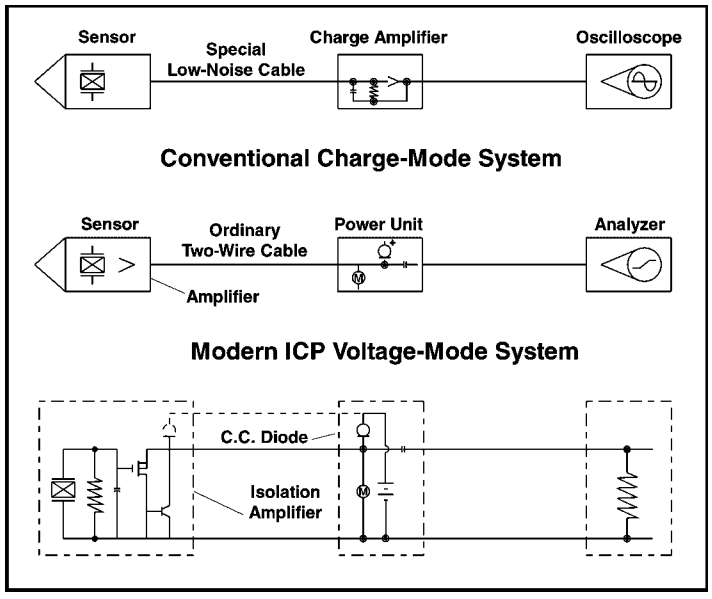


FIGURE 5
COMPARISON OF CHARGE MODE AND ICP VOLTAGE-MODE ACCELEROMETERS (REF. 3)

© Copyright 2001 Technical Associates of Charlotte, P.C.

Therefore, the most commonly used PMP accelerometer today is the low impedance, ICP type containing the built-in microelectronics which condition the signal from a crystal within the accelerometer to a low impedance voltage which is compatible with the readout device. A constant current power source is required for the transistor within the accelerometer to turn on and perform its measurement (again this constant current power source is normally available in most all FFT analyzers today). Advantages of the ICP accelerometer include fixed sensitivity, ability to operate reliably in dirty and moist environments, the use of ordinary signal transmission cables, the lack of requirement of a separate signal conditioner if one is available in the analyzer and typically a lower system cost. Disadvantages sometimes include limited temperature range due to the survivability of the built-in electronics (although this can sometimes be overcome by use of special ceramic construction materials). Another disadvantage which is important to point out is that if an ICP-type accelerometer is not hermetically sealed, the entry of moisture into the transducer can significantly degrade its low frequency performance. Reference 4 points out that:

> "All accelerometers breathe, unless they are certified as hermetically sealed, and during this micro-breathing, moisture inevitably penetrates. As already mentioned, the low-frequency cutoff in a voltage amplifier depends on the accelerometer as well as cable resistance and capacitance. As moisture penetrates, the internal resistance decreases from the gigaohm range to the megaohm range. This changes amplifier performance dramatically. It has been observed that some transducers with built-in preamplifiers gave no useful signal below 40 Hz after only a few months of use on paper making machinery"[4]

Therefore, it is important that if one acquires an ICP-type accelerometer, acquire a hermetically sealed unit, particularly if it is to be used in humid environments. This is especially important in those cases where low-frequency machines are to be evaluated.

One final important point needs to be considered when choosing accelerometers. It is important that the user be aware of the phase response of the particular transducer and how it varies with frequency. This is illustrated by Figure 6 which shows an accelerometer with fairly constant phase response from approximately 5% up to about 60% of its mounted resonance frequency (f_m). Note in Figure 6 that as the transducer approaches its mounted resonance, it begins to experience a

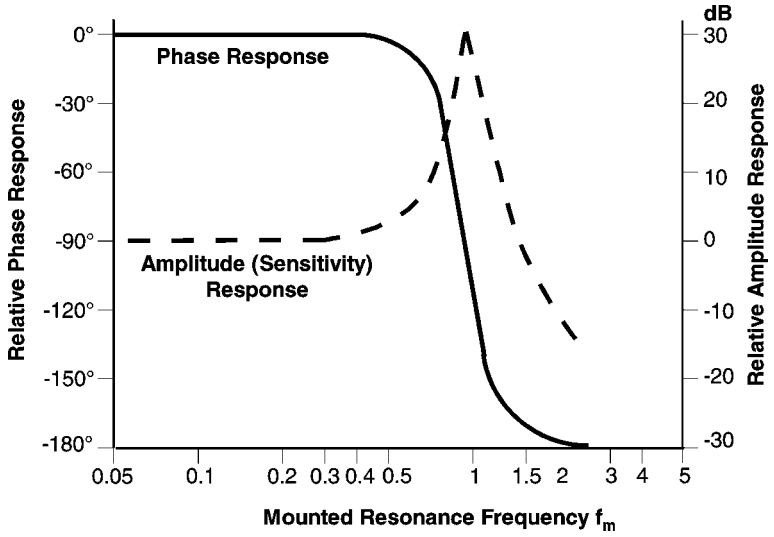


FIGURE 6
SCHEMATIC OF ACCELEROMETER SENSITIVITY AND PHASE RESPONSE AS A FUNCTION OF FREQUENCY (REF. 7)

© Copyright 2001 Technical Associates of Charlotte, P.C.

dramatic phase change which approaches almost 180° when it passes through this resonance. Therefore, if one were using this accelerometer when performing a phase analysis on a machine, the user must be aware of the frequency range at which the phase response of the transducer itself undergoes this significant change. This can totally negate the accuracy of the phase analysis which would be important in helping clearly identify whether high vibration is caused by symptoms having similar vibration signatures such as unbalance, misalignment, bent shaft, eccentricity, etc. One must also be aware that if a magnet or an extension probe is used to mount this same accelerometer, this will significantly lower the natural frequency, and therefore, will significantly lower the frequency ranges in which accurate phase measurements can be made with this same accelerometer.

Note that the **APPENDIX** includes examples of many different accelerometers from various vendors, including their specifications. Comparing these, note that a very wide range of accelerometers exist ranging from those capable of measuring down to as low as .01 Hz (.6 CPM) up to over 60,000 Hz (3,600,000 CPM). In addition, note that voltage sensitivities will vary from as low as about 0.4 mV/g up to as high as 10,000 mV/g (10 V/g); and that accelerometer masses can vary from as low as 1 gram up to over 1000 grams. Therefore, in most predictive maintenance programs which are outfitted not only with normal speed machinery ranging from about 600 up to 3600 RPM, but also with low speed (particularly below 200 RPM) as well as high speed machinery (particularly above 10,000 RPM), it will very likely be necessary to have at least three different accelerometers - a low frequency, a general purpose and a high frequency transducer. Very likely, the general purpose accelerometer can probably be used to successfully make over 90% of the measurements. However, the accuracy of measurements on the very low or very high speed machines will suffer without the low and high frequency accelerometers (and therefore, results will prove far less effective).

4.12 VELOCITY PICKUPS

As of today, velocity pickups have generally been replaced by accelerometers for most applications because of their expense, limited frequency range, and relatively heavier weight. However, one advantage of velocity pickups is that they do not have to be powered by an external power supply such as ICP current or a separate charge amplifier. Also, since velocity is the commonly accepted unit of vibration analysis for most PMP programs, their signal does not have to be integrated from acceleration to velocity (as is the case with accelerometers). Therefore, this eliminates integration noise which typically occurs in the first 2 to 4 lines of the FFT spectra taken by accelerometers integrated to velocity.

There are two types of velocity pickups - seismic velocity transducers and piezoelectric velocity pickups. Figure 7 shows an example seismic velocity transducer while Figures 9 and 10 show example piezoelectric velocity pickups.

Unlike the piezoelectric velocity transducers (and the piezoelectric accelerometers previously discussed), the seismic velocity pickup is self-generating and does not require an external power supply of any kind. Figure 7 shows that the seismic velocity pickup consists of a case in which a coil of wire is wound around a seismic mass which itself is supported by soft springs. This gives the unit a very low natural frequency on the order of only 10 Hz (600 CPM). In addition, a permanent magnet is firmly attached to the transducer case providing a strong magnetic field around the coil. In essence, whenever the velocity pickup is mounted on a bearing housing, the permanent magnet will vibrate while the spring-suspended coil of wire will remain stationary in space. When the moving magnetic field cuts through the stationary coil of wire, a voltage proportional to the relative velocity of the magnet and coil is generated, with the voltage determined by the number of turns of wire in the coil. The voltage generated is then transmitted out of the transducer into a vibration analyzer.

© Copyright 2001 Technical Associates of Charlotte, P.C.

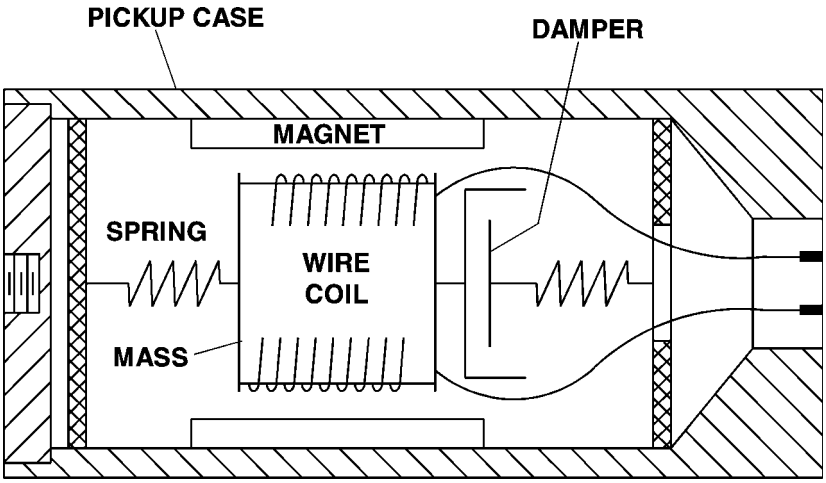


FIGURE 7
IRD 544 SEISMIC VELOCITY PICKUP (Ref. 5)

Frequency Range (± 10%):	600-60,000 CPM (10-1000 Hz)
Sensitivity (± 10%):	764 mV RMS / in/sec PK = 1080 mV Peak / in/sec PK
Temperature Range:	-40° F to + 500° F (-40° C to + 260° C)
Impedance:	R = 2000 ohms; L= 0.7 Hy
Damping Factor:	0.7 (critical)
Transverse Response (Avg):	Less than 5% (10-1000 Hz)
Magnetic Field Sensitivity:	0.15 in/sec /Gauss
Weight:	21oz (595 grams)

The suspended mass shown on Figure 7 is damped (either electrically or with synthetic oil) such that it remains motionless throughout the frequency range as the case vibrates around it. Therefore, the transducer's low frequency response is limited by its first natural frequency, with the sensitivity rolling off typically around 600 to 1200 CPM, (or 10 to 20 Hz). Therefore, this sensitivity will drop as the mass overcomes the damping effect at low frequencies and begins to move in phase with the vibration. Figure 8 shows what must be done if measurements are required by an example velocity pickup (IRD 544) if measurements are desired below 600 CPM (10 Hz). For example, if a measurement of .1 in/sec were recorded using this transducer at 300 CPM, Figure 8 shows that the actual level would be about 3.3 times higher than this (or about .33 in/sec). Note that with decreasing frequencies, significantly higher and higher multiplication factors must be applied to correct the vibration readings. Therefore, a seismic velocity pickup is not a good low frequency transducer.

In addition, the upper frequency response is limited on most velocity pickups to no more than about 60,000 to 120,000 CPM (1000 - 2000 Hz) due to the increasing difficulty of overcoming system inertia. Another problem with these units is that they typically are very large in size and weight [the example in Figure 7 weighs 21 ounces (or 595 grams)], which can undesirably contribute to mass loading of the machine or structure being evaluated.

If much lower or higher frequency measurements with a direct velocity readout are required, a piezoelectric velocity transducer like that shown in either Figure 9 or Figure 10 is recommended. In these cases, Figure 9 shows again that a piezoelectric disk is mounted below a seismic mass,

© Copyright 2001 Technical Associates of Charlotte, P.C.

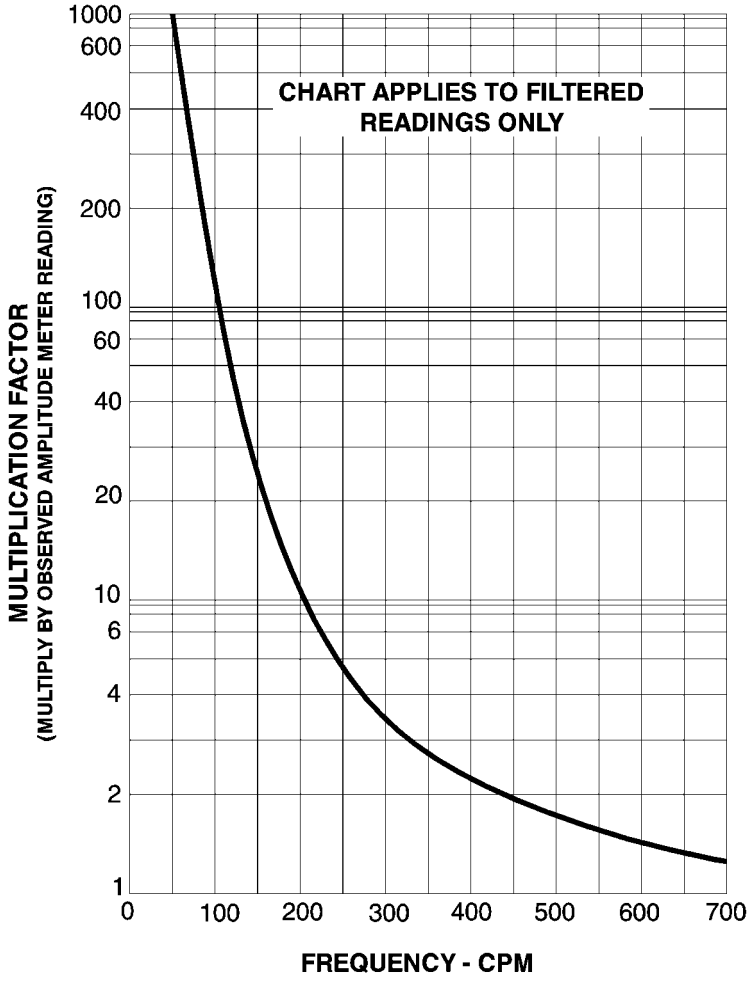


FIGURE 8
CORRECTION FACTOR CHART FOR IRD 544 SEISMIC VELOCITY PICKUP WHEN MEASURING FREQUENCIES BELOW 600 CPM (10 Hz) (REF. 5)

© Copyright 2001 Technical Associates of Charlotte, P.C.

along with an internal analog integrator which converts the acceleration signal to velocity. This analog integration inside the transducer case will sometimes decrease the noise at low frequencies due to integration and other sources. This piezoelectric crystal also provides the unit a small amount of phase shift making it a much better transducer for phase analysis and balancing than the seismic velocity pickup. In addition, one of the problems with seismic transducers is that they are most often highly susceptible to magnetic interference. On the other hand, piezoelectric velocity pickups and piezoelectric accelerometers are virtually unaffected by magnetic fields (see comparative magnetic field sensitivities in Figures 7 and 9, respectively).

Finally, the frequency range of piezoelectric velocity pickups is much wider than that of seismic units as seen by comparing Figure 7 (seismic) with Figures 9 and 10 (piezoelectric). Note that the unit of Figure 9 has a frequency response (±10%) of 60 up to 270,000 CPM (1 - 4500 Hz) while that of Figure 10 can measure as low as 150 CPM (2.5 Hz) up to 210,000 CPM (3500 Hz). Further examples of similar velocity transducers, along with their specifications, are contained within the **APPENDIX**.

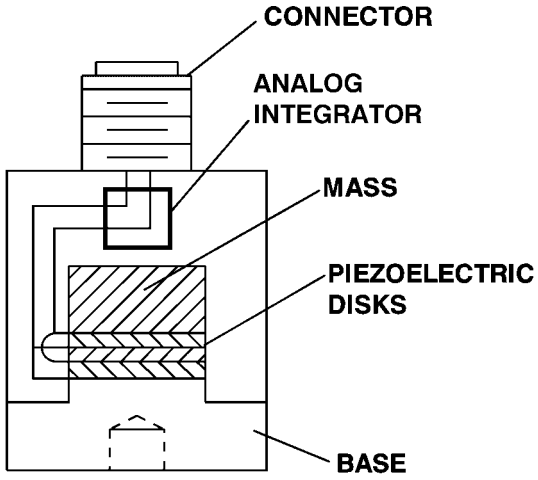


FIGURE 9
IRD 560 PIEZOELECTRIC VELOCITY PICKUP (Ref. 5)

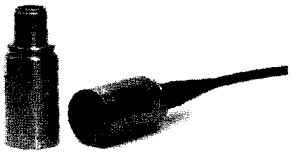
Frequency Range (± 10%):	60-270,000 CPM (1-4500 Hz)
Sensitivity (± 10%):	764 mV RMS / in/sec PK
	= 1080 mV Peak / in/sec PK
Temperature Range:	-40° F to + 250° F
	(-40° C to + 121° C)
Mounted Natural Freq.:	600,000 CPM min. (10,000 Hz)
Transverse Response :	5% max @ 600 CPM (10Hz), 5 g peak
Magnetic Field Sensitivity:	.00001 in/sec /Gauss
Weight:	38 oz (1079 grams)

© Copyright 2001 Technical Associates of Charlotte, P.C.

WILCOXON RESEARCH

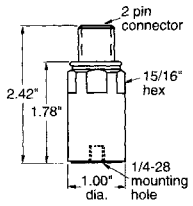
INDUSTRIAL

793V Series
Piezoelectric Velocity Transducers

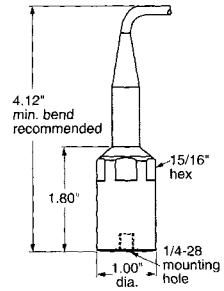


FEATURES:

- Intrinsic safety certified option
- Industrial ruggedness
- Eliminates distortion caused by high frequency signals
- Corrosion-resistant
- Internally integrated to velocity
- Ultra low-noise electronics for clear signals at very low vibration levels
- Miswiring protection



793V Series Models



793VS Series Models

SPECIFICATIONS	All Models	793V	793V-5	Units
DYNAMIC				
Sensitivity, ±10%, 25°C	—	100	500	mV/in/sec
Velocity Range	—	50	10	in/sec peak
Amplitude Nonlinearity	1%			
Frequency Response:				
±10%	—	2.0 - 3,500	8.0 - 3,500	Hz
±3 dB	—	1.5 - 7,000	5.0 - 7,000	Hz
Resonance Frequency, mounted, nom.	15 kHz			
Transverse Sensitivity, max.	5 % of axial			
Temperature Response	see graph			
ELECTRICAL				
Power Requirement:				
voltage source	18 - 30 VDC			
current regulating diode[1]	2 - 10 mA			
Electrical Noise, equiv. in/sec, nominal:				
Broadband 2.5 Hz to 25 kHz	—	100	40	µin/sec
Spectral 10 Hz	—	10	4.0	µin/sec/√Hz
100 Hz	—	1.0	0.4	µin/sec/√Hz
1000 Hz	—	0.2	0.05	µin/sec/√Hz
Absolute Phase Shift, nom. the greater of:	—	$\tan^{-1}$ 2/f or 2°	$\tan^{-1}$ 5/f or 2°	
Output Impedance, nominal 4mA supply, the greater of:	—	5,000/f	25,000/f	
	—	or 200Ω	or 200Ω	
Bias Output Voltage, nominal	10 VDC			
Grounding	case isolated, internally shielded			
ENVIRONMENTAL				
Temperature Range	–50 to 120°C			
Vibration Limit	250 g peak			
Shock Limit	2,500 g peak			
Electromagnetic Sensitivity, equiv. in/sec	25 µin/sec/gauss			
Sealing	hermetic			
Base Strain Sensitivity	0.0005 in/sec/µstrain			
PHYSICAL				
Weight	145 grams			
Case Material	316L stainless steel			
Mounting	¼ - 28 tapped hole			
Output Connector	MIL-C-5015 2-pin			
Connections Pin A signal, power				
Pin B common				
Cabling[2]: Mating Connector	Wilcoxon R6 (Amphenol 97-3106A-10SL-4S)			
Recommended Cable	J9T2, two conductor shielded, Tefzel jacket, 30 pF/ft.			

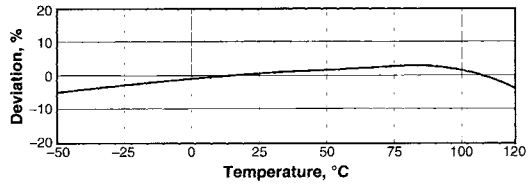
NOTES: [1] A maximum current of 6 mA is recommended for operating temperatures in excess of 100 °C.
[2] Cabling for "S" option model is J61, coaxial, polyurethane jacket, 30 pF/ft.

OPTIONS: Intrinsic safety certification; stainless steel flexible hose; 10 ft. sealed integral cable "S" or "SG", temperature sensor "T", customer specified sensitivity, low frequency filtering, cable length, connector.

ACCESSORIES SUPPLIED: SF6 mounting stud, calibration data, CCD card

ACCESSORIES AVAILABLE: R6SL-J9T2-XX splash-proof cable assembly, magnetic mounting bases, SF8 cementing studs, power supplies, amplifiers, signal conditioners.

TYPICAL TEMPERATURE RESPONSE



TYPICAL FREQUENCY RESPONSE

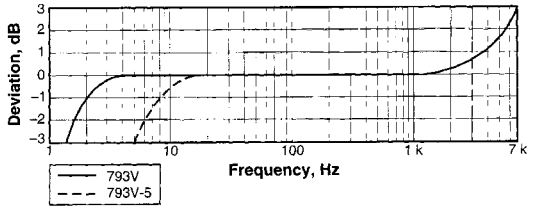


FIGURE 10
793V SERIES PIEZOELECTRIC VELOCITY TRANSDUCERS (Ref. 5)

© Copyright 2001 Technical Associates of Charlotte, P.C.

4.13 NON-CONTACT EDDY CURRENT DISPLACEMENT PROBES

Figure 11 pictures a non-contact eddy current displacement probe system (sometimes called proximity probes). These systems are primarily used to measure shaft vibration, radial and axial shaft position and differential expansion between the case and rotor. They provide direct relative shaft vibration (shaft vibration relative to the non-contact mounting point on the machine housing). On the other hand, accelerometers and velocity pickups mounted on the machine housing measure absolute vibration. Eddy Current Probes are especially effective on large machinery with oil film plain bearings such as turbine/generators, compressors, large motors, etc. In these cases, there is often a significant signal loss by the time the shaft vibration passes through the oil film to the bearing, and then through potentially several metal interfaces out to the machine housing, often at distances of 2 feet or more. In these cases, shaft vibration can be 10 to 20 times higher than that for smaller machines with rolling element bearings. Therefore, the best possible vibration spectrum, particularly for events at 1X RPM up through about 6X RPM is that which can be acquired directly from a non-contact eddy current probe. This probe will measure vibration of the shaft relative to the mounting system of the probe itself (therefore, it is most important to anchor the probe as much as possible to get the truest picture of shaft vibration).

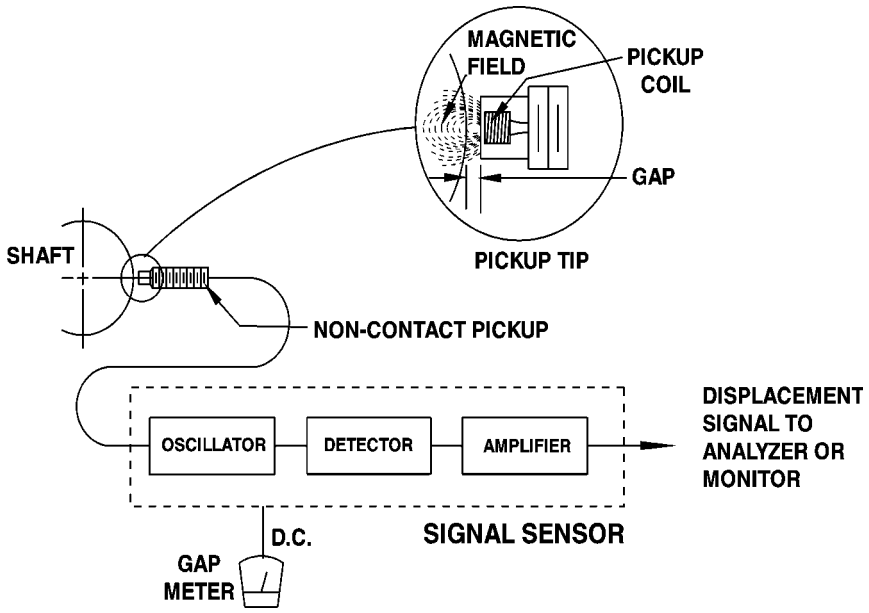


FIGURE 11
SCHEMATIC DIAGRAM OF AN EDDY CURRENT NON-CONTACT DISPLACEMENT PROBE SYSTEM (REF. 5)

Figure 11 shows a schematic diagram of an eddy current non-contact probe system. Unlike piezoelectric accelerometers and velocity pickups, a non-contact transducer does not generate a voltage or electrical charge in response to vibration. Instead, a non-contact sensor requires external electronic circuitry to generate a very high frequency A.C. signal and detects oscillations in the A.C. signal caused by vibration of the shaft. This high frequency electrical signal is called the carrier signal, which is applied through a coaxial cable to a small coil of wire in the tip of the non-contact pickup. Figure 11 shows that this high frequency electrical signal applied to the coil which generates a magnetic field. Then the target surface, which will often be placed at a gap of

© Copyright 2001 Technical Associates of Charlotte, P.C.

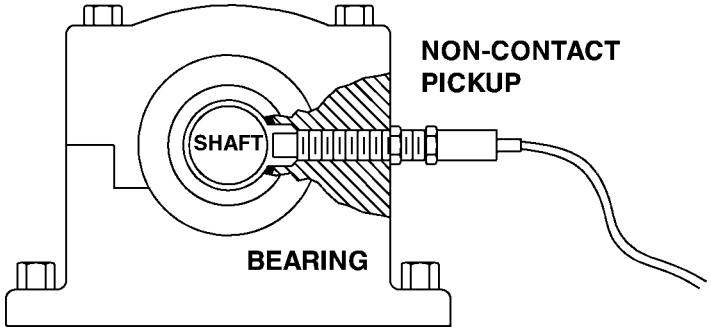


FIGURE 12
NON-CONTACT EDDY CURRENT PROBE MOUNTED IN A BEARING HOUSING (REF. 5)

only 40 to 60 mils from the non-contact probe tip, will absorb some of the magnetic energy with the resulting absorption in effect placing an electrical load on the signal reducing its strength. Therefore, as the shaft moves relative to the pickup tip, the strength of the electrical signal changes proportional to the movement. The probe will provide an A.C. signal voltage proportional to the vibration and a D.C. signal proportional to the gap. Thus, the A.C. signal can be read by a permanent monitor or an analyzer to determine the amount of shaft vibration while the D.C. signal can be used to monitor the change in gap (particularly useful if targeted towards the shaft end to detect axial shaft movement and possible wear of bearings). Figure 12 shows a non-contact probe mounted in a bearing housing. Here again, the probe itself will measure the shaft displacement relative to the bearing and housing. Again, the signal from the probe can be taken directly into an analyzer using output connectors from the transducer. In these cases, be very careful! Do not connect the cable to the ICP port on a spectrum analyzer or data collector because the ICP current will cause the vibration readout to suddenly jump which can cause these machines to alarm, or even worse, trip off line.

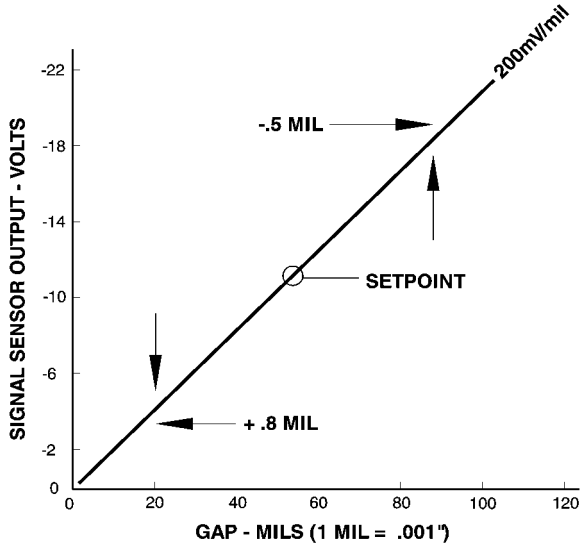


FIGURE 13
TYPICAL OUTPUT CURVE FOR AN EDDY CURRENT NON-CONTACT DISPLACEMENT PROBE SYSTEM

© Copyright 2001 Technical Associates of Charlotte, P.C.

Figure 13 shows a typical response curve for a non-contact probe and signal sensor. This curve represents the change in sensor output voltage as the distance between the tip of the pickup and the target material is changed. Looking closely at Figure 13, note that the relationship of the output voltage to the gap approximates a straight line. The slope of this line, millivolts per mil (mV/mil) determines the non-contact probe sensitivity. In general, the linearity of this system will be defined as that portion of the output voltage versus gap within which the slope does not vary by more than ±5%, or for which the absolute gap error does not exceed ±1.5 mils. In the system pictured in Figure 13, this system has a linear gap range from about 20 to 100 mils or a total of 80 mils. Therefore, the optimum " setpoint" for this non-contact pickup system would be selected at the midpoint (60 mils) of the specified linear gap range.

Displacement Probe System

Of great importance when installing non-contact probes is providing a very smooth, uniform surface finish on the target. In fact, a common problem with non-contact probes is that the target material must be free of any surface imperfections such as scratches or high spots. When a scratch is present, the probe cannot discern between true vibration and the scratch depth, thereby causing an error in the readout. However, if one examines time waveforms and/or vibration spectra from a shaft having a scratch, it will be very noticeable such as the time waveform caused by a scratch on a target shown in Figure 14.

Another problem common to non-contact probe systems is runout - either mechanical or electrical runout. Mechanical runout is an error in measuring the position of the shaft center line with a displacement probe which is caused by out-of-roundness and surface imperfections (scratches, chain marks, dents, rust or other conductive build-up on the shaft, flat spots and engravings). It is most important to ensure that there is less mechanical runout in the shaft than the minimum vibration displacement which is to be measured. Readings should be taken at 30° - 45° rotational intervals around the shaft plotting runout versus angular position to ensure this and also to provide a baseline for comparison with electrical runout data.

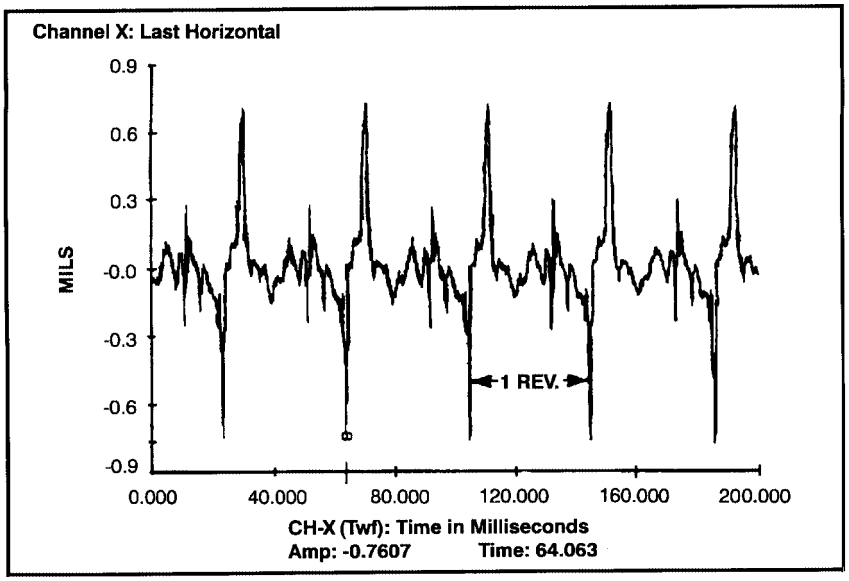


FIGURE 14
TIME WAVEFORM TAKEN FROM A NON-CONTACT PROBE TARGETED ON A SCRATCHED SHAFT (REF. 10)

© Copyright 2001 Technical Associates of Charlotte, P.C.

Electrical runout is a signal error which occurs in eddy current displacement measurements when the shaft surface conductivity varies. That is, the target's surface itself has non-uniform electrical conductivity/resistivity/permeability properties. Another source of electrical runout is due to the presence of a local (spot) magnetic field(s) at a point(s) on the shaft surface. This error repeats exactly with each shaft revolution. Such shaft magnetism may result from magnetic-particle testing or simply from operating in a strong magnetic field which sometimes occurs with motors, generators or alternators. In addition, welding operations conducted nearby a machine can sometimes induce electrical runout into a shaft. Here again, it is impossible for the displacement probe to differentiate true vibration from either the mechanical or the electrical runout, depending on where exactly the electrical and mechanical runout is located on the shaft, the runout may either add or subtract from one another and from the total vibration. That is, the displacement readout will be a vectorial addition of electrical runout, mechanical runout and actual vibration. There are procedures in place for determining the presence of mechanical and electrical runout and for minimizing its effect. In the case of electrical runout, it often is necessary to correct this with a degaussing instrument.

It is very important that the non-contact probe be calibrated for the specific shaft material to which it will be targeted. Figure 15 shows typical response curves for many different target materials as well as comparing it to the gap set between the probe and the target. This chart is used to set up and calibrate the response of the proximity probe. It is important to notice that "roll-off" problems occur as the gap is increased. It is not recommended that probes be used in this roll-off range.

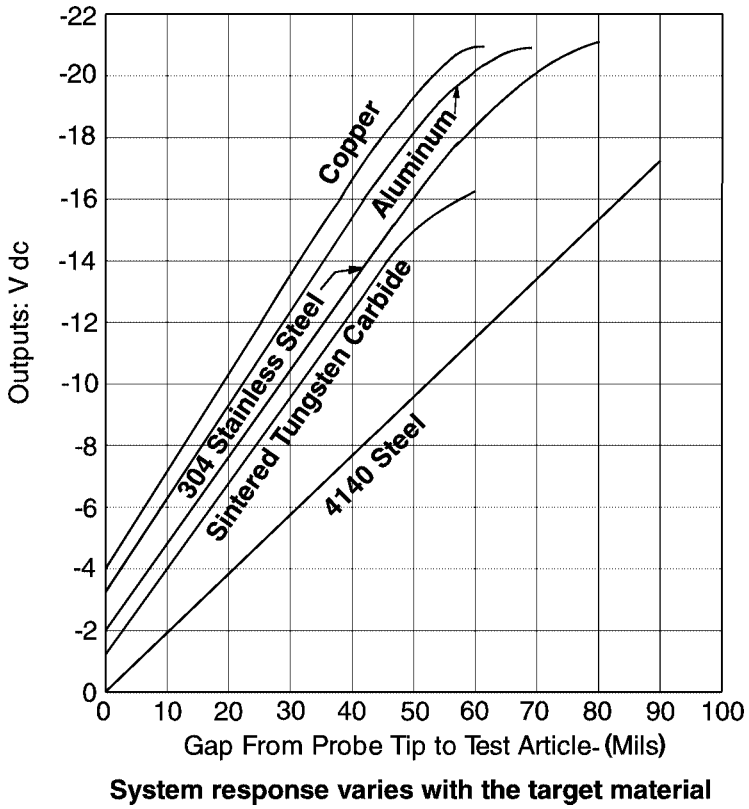


FIGURE 15
VOLTAGE RESPONSE VERSUS GAP FROM PROBE TIP TO SHAFT FOR VARIOUS TARGET MATERIALS (REF. 8)

© Copyright 2001 Technical Associates of Charlotte, P.C.

Finally, one of the remaining problems with a non-contact probe is frequency range. Typically, specifications for non-contact probe systems will claim a frequency response of from about 0 to 600,000 CPM (0 - 10,000 Hz). However, this does not actually occur in the real world due to practical reasons. For example, in the real world, since the probe is measuring displacement, it will tend to emphasize low frequency sources, but de-emphasize high frequency sources. For example, consider a centrifugal compressor outfitted with non-contact probes on each of its impellers which range in operating speeds from about 20,000 up to approximately 50,000 RPM. For all practical purposes, since the displacement amplitudes become so low with increasing frequencies, these displacement measuring systems will not be sufficiently sensitive to pick up problems with the gearing or with failing thrust bearings on these machines (which are commonly outfitted with about 6 pads in each thrust bearing). However, if problems with unbalance due to impeller buildup and/or erosion occur, or if a misalignment problem takes place, the system will readily respond whether it is targeting the low-speed or high-speed impeller. But, if, for example, a problem occurs with a thrust bearing of either stage, the displacement probe will not likely report any real problem since the thrust bearing frequencies will range from 120,000 CPM up to 300,000 CPM. In the case of these machines, a common alarm level is at about 0.7 mil (Pk-Pk) while a normal trip set point is at about 1.1 mil (Pk-Pk).

Therefore, calculations show that if the low-speed stage had a high velocity of 2 in/sec Pk shaft vibration at the 120,000 CPM frequency, this would correspond to an equivalent displacement of only .318 mil (Pk-Pk) (and only .127 mil at 300,000 CPM on the high-speed thrust bearing). Even worse, these systems will not be responsive to wear of the pinion teeth since the typical fundamental gear mesh frequency (GMF) for these machines is on the order of 1,000,000 CPM (16,667 Hz). Therefore, calculations show that an excessive vibration of 100 g's acceleration at this fundamental gear mesh frequency of about 1,000,000 CPM would correspond to an equivalent displacement of only .0071 mil (Pk-Pk), making it impossible for the system to detect this pinion tooth wear. Still, however, these machines should always be outfitted with such non-contact probe permanent vibration monitors in order to detect the actual shaft vibration and the onset of problems such as misalignment and unbalance to which they are, in fact, sensitive.

4.14 SHAFT CONTACT DISPLACEMENT PROBES

Shaft contact probes (shaft sticks and shaft riders) are probes that actually ride on the shaft surface of the machine to measure the vibration. They are an older technology which has generally been replaced by the non-contact eddy current probe in most cases. However, they are still in use for specific applications such as balancing or periodic vibration checks if a proximity probe is not installed and a direct measure of shaft vibration is required. These transducers are very limited in frequency response and are not capable of measuring vibration much above 12,000 CPM (200 Hz). Also, in general, they should not be used when shaft surface speeds exceed approximately 30,000 ft/min (±10,000 m/min).

4.141 Shaft Sticks

Figure 16 is a picture of a shaft stick in use. It typically consists of a hardwood, fish-tailed" stick outfitted with a stud on its other end for attachment of a velocity pickup or accelerometer transducer. The fish-tail shape provides two points of contact required to help the shaft stick in contact with the shaft throughout its circumference and also allows it to be used with a variety of shaft diameters. Also, it is a good idea to taper the end of the stick to provide a minimum contact area with the shaft to reduce friction and to prevent chatter with the shaft. The shaft stick is hand-held to the shaft. Figure 17 shows the use of a shaft stick to measure both vibration amplitude and phase to confirm a bent shaft condition with readings taken on the shaft on each side of the bearing (if the shaft is bent, phase readings will likely differ by approximately 180° from that taken on the other side of the bearing).

© Copyright 2001 Technical Associates of Charlotte, P.C.

As with the non-contact eddy current probe, the shaft stick is often used in applications where journal bearings are installed or where the housing or casing is very heavy compared to the shaft or rotor and vibrations are absorbed by the shaft clearances, lubrication, or heavy casings. However, the shaft stick has a much lower usable frequency range than does a non-contact probe due to its construction, and typically should only be used if a non-contact eddy current probe is not installed.

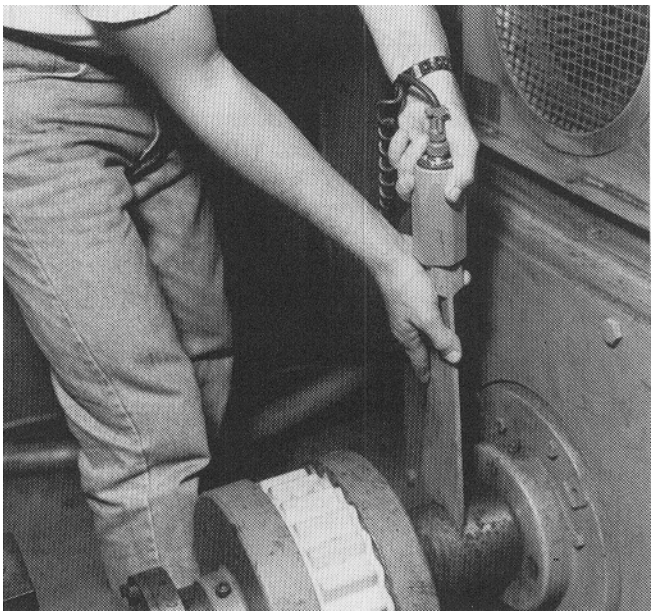


FIGURE 16
SHAFT STICK (REF. 5)

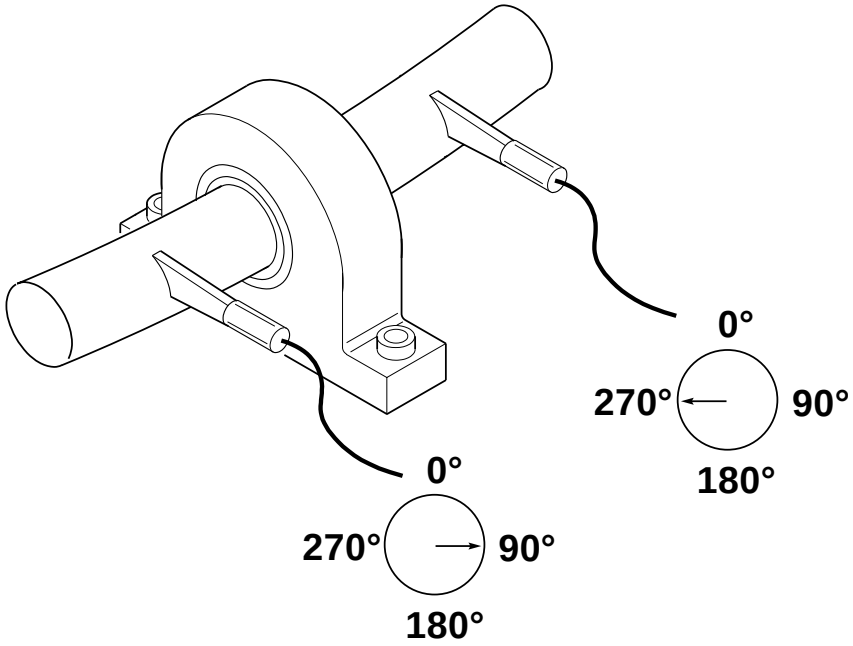


FIGURE 17
PHASE READINGS USING SHAFT STICK ON BOTH SIDES OF BEARING HELP IN CONFIRMING A BENT SHAFT

© Copyright 2001 Technical Associates of Charlotte, P.C.

When using the shaft stick, use special care to follow these precautions:

- Hold the stick tightly to prevent it from “walking down” the shaft.
- Keep the stick perpendicular to the shaft.
- Do not mount the stick on shafts traveling faster than 10,000 RPM for safety reasons.
- Do not keep the stick on the shaft for extended periods of time which may burn the stick or score the shaft.
- Apply a continuous stream of lubricant to the shaft while taking the measurement to reduce friction, burning and scoring.
- Try to use the same pressure when taking PMP type measurements as this will affect your month to month vibration amplitudes.
- Make sure the shaft is finished, smooth and free of rust, pits or dents which can produce useless information as well as damage the stick.
- Do not rely on the shaft stick to monitor “high frequency” defects such as bearings, blade passes, gear mesh frequencies, and electrical problems.

4.142 Shaft Riders

A shaft rider (pictured in Figure 18) is similar to the non-contact eddy current probe in that it is permanently mounted to the housing and extends into the casing to measure shaft vibration displacement. However, it differs from the proximity probe in that it actually rides on the shaft surface by typically using a spring mounted tip. On the other end of the shaft rider, a transducer is mounted. One of two methods of picking up the vibration is usually used. The first has an accelerometer or velocity transducer mounted on the end of the shaft rider measuring the vibration (this case is quite similar to using a probe with an accelerometer or velocity transducer). The second employs a method in which a spring loaded rod is driven by the vibration and moves a mass inside of coiled wires surrounded by magnets, thereby generating an electrical signal which is used to measure the vibration.

Once again, the shaft rider is an older technology which has gradually been replaced by the non-contact eddy current probe. Typically, the shaft rider was employed as a device to continually monitor vibration of the shaft for purposes of determining if a machine’s vibration. level should sound an alarm or shut down the machine. As with the shaft stick, its usable frequency range is up to about 11,000 CPM which is only useful for monitoring unbalance, misalignment, and other problems that occur at 1X or 2X RPM. Furthermore, the shaft speed must not exceed 3600 RPM to be able to detect these problems and should be used only on machines with a shaft surface speed below approximately 20,000 ft/min (6100 m/min). The shaft rider will not be able to detect gear mesh problems and other high frequency problems.

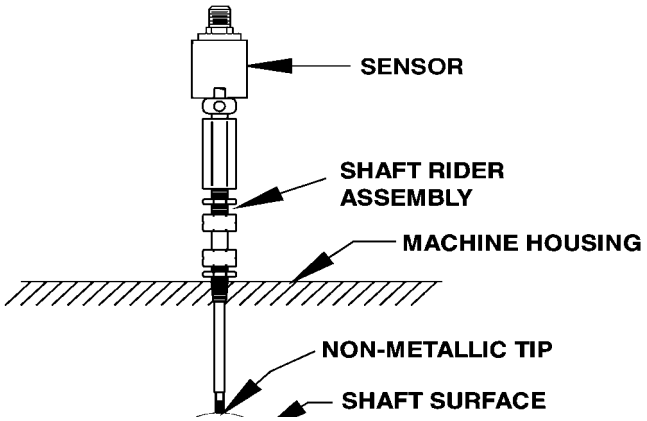


FIGURE 18
SHAFT RIDER

© Copyright 2001 Technical Associates of Charlotte, P.C.

4.2 SELECTION CRITERIA FOR TRANSDUCERS

In order to help the reader better understand vibration transducers, a list of common transducer characteristics is presented in **TABLE I**, along with a summary of each type of transducer's typical capabilities. Table I, along with the information shared in Section 4.1 will help the analyst choose the right transducer for a particular job. Following below are descriptions of the items in Table I, along with other items which should be considered when selecting a transducer:

1) TYPICAL SENSITIVITY RANGE - Sensitivity is the capability of the transducer to determine the amplitude of vibratory motion (displacement, velocity, or acceleration) from the amplitude of the voltage signal. An example might be 100 mV/g (for an accelerometer), where "g" is 1 gravity unit (32.2 ft/sec^2 or 386 in/sec^2). For example, if this 100 mV/g accelerometer generated 10 mV, it would convert this voltage to an amplitude of 0.1 g (10/100 = 0.1).

2) TYPICAL FREQUENCY RANGE - Frequency range is the low and high end frequency measuring capability of the transducer. Each individual transducer has its own frequency range (which must be known by the user). However, each type of transducer has a relative range that is displayed in **TABLE I.** The low end of the frequency range is primarily controlled by the sensitivity of the transducer where the high end frequency limit is controlled by the transducer's mounted natural frequency. Typically, the transducer's frequency response might be given for various amplitude tolerances such as ±5% ±10%, and/or ±3dB (for example, see Figure 10).

3) NATURAL FREQUENCY RANGE - The natural frequency of the transducer is the limiting factor for the upper frequency range capability of the transducer. When a forcing frequency begins to approach the transducer natural frequency, the amplitude of vibration will be magnified up to as much as 50 times due to the resonant condition of the transducer. Therefore, it is very important that the highest frequency of interest is well below the natural frequency of the transducer. Normally, the effective accurate range for an accelerometer (±5%) will be about 20% to 33% of its stud mounted natural frequency (thus, for an accelerometer with a 30,000 Hz natural frequency, its effective range will likely be limited to only about 6000 to 10,000 Hz, or 360,000 to 600,000 CPM).

4) TYPICAL WEIGHT RANGE - The weight of the transducer is important for the following two reasons. First, it must be light enough to carry around on a PMP route. Second, the transducer weight must only be a fraction of the weight of the housing to which it will be attached. If the transducer is heavy enough to affect the mass of the part being measured, it can have a significant effect on the part's natural frequency, and therefore, true frequency response.

5) TYPICAL USABLE TEMPERATURE RANGE - This is the minimum and maximum temperature that a transducer can withstand without significantly affecting its response capabilities. Make special note of this parameter when attaching permanent transducers inside of machinery that may have very high or very low temperatures.

6) MEASUREMENT DIRECTION - Most transducers measure only in the mounting direction (although they may respond to about 3% to 5% of the vibration from directions perpendicular to the orientation of the transducer). However, a few allow vibration to be collected in all three directions simultaneously (triaxial transducers).

7) TRANSDUCER SIZE - Some of the measurement locations on machinery require that the transducer be of a certain size or cross section to physically fit on a small surface for instance. Therefore, a small transducer must be used. However, due to the small size, the transducer may have a low voltage sensitivity restricting its capability to pick up low frequency information. Also, if the object that is vibrating is very small, the added mass of the transducer may change the natural frequency of the object, thereby changing its vibratory motion.

© Copyright 2001 Technical Associates of Charlotte, P.C.

8) TRANSDUCER POWER SUPPLY - Most transducers require a power supply to operate. There are 3 common methods to power a transducer:

i) ICP (Integrated Circuit Piezoelectric) Power Supply. The vibration instrument sends the power through the cable attaching it to the transducer to power the transducer.

ii) Independent Power Supplies. A battery or AC powered unit is used to send power to the transducer.

iii) Charge Amplifier. A battery or AC powered unit is placed between the transducer and the vibration instrument to amplify the signal voltage.

9) CABLING - Cable length should be considered when purchasing your transducers. ICP powered transducers can generally use cables up to about 1000 feet in length. If the transducer is not ICP powered, a charge amplifier may be required at approximately 50 foot intervals of cable length (or less). Discuss the cabling requirements for the transducer with the transducer vendor.

10) MOUNTING SENSITIVITY - As will be described in **Sections 4.3 and 8**, there are many ways to mount transducers (hand- held probes, magnetic connectors, permanent stud mounts, etc.) and each has a significant effect on the ability of the transducer to measure the vibration accurately, as well as to reproduce the same spectra in subsequent measurements. This one fact is often critical to obtain accurate, repeatable data (and, therefore, to the success or failure of the entire PMP program).

11) MAGNETIC INTERFERENCE - Magnetic interference may affect the performance of the transducer, cable, and vibration instrument. Proximity probe type transducers have a magnetic field to determine the shaft gap from the transducer. If another magnetic field (introduced by electrical runout for instance) interferes with the proximity probe's field, erroneous vibration readings will be recorded. Typically, magnetic interference does not affect transducers with piezoelectric crystals. However, it can affect transducers which work on a spring-mass system (such as seismic velocity pickups). Also, cable movement generates a magnetic field and can distort the signal in the cable. Lastly, large generators or motors can produce a magnetic field which may affect spring-mass transducers, cables, and the vibration analyzer itself.

12) SEALING METHOD - Remember, all accelerometers will "breathe" if they are not certified as hermetically sealed. During this micro-breathing, moisture will inevitably penetrate those units which are not hermetically sealed which will change the transducer's performance dramatically, particularly at lower frequencies below approximately 3000 CPM (50 Hz). Pay attention to the specifications for your particular accelerometer. If its sealing is not listed, contact your vendor and ask this important question.

© Copyright 2001 Technical Associates of Charlotte, P.C.

4.3 MOUNTING OF TRANSDUCERS

Typically, there are 5 different transducer mounting methods as listed in Table II and each has a typical frequency response range when used with transducers capable of these frequency response ranges.

TABLE II
TRANSDUCER MOUNT USABLE FREQUENCY RANGE
FOR THE WILCOXON 726T (Ref. 9)

ACCELEROMETER MOUNTING	MAXIMUM ACCEPTABLE FREQUENCY (CPM)	MOUNTING NATURAL FREQUENCY (CPM)
1) Stud Mount	975,000	1,900,000
2) Adhesive Mount with Hottinger Baldwin Messtechnik X60	540,000	None Observed
3) Stud Mount on Rare Earth Magnet	450,000	724,500
4) Mounted on Quick-Connect Stud Mount	360,000	609,000
5) Hand-held Mount Using a 2" Probe	48,000	88,500

Table II is a summary of an article written by Computational Systems, Inc. (CSI)[9]. Many other factors come into play when the effectiveness of a transducer mounting type is considered. Refer to the referenced article for an in-depth look at transducer mounting effectiveness. Figure 19 shows illustrations of these mounting methods (Quick-Lock not shown).

4.31 TRANSDUCER MOUNTING APPLICATIONS

Each particular transducer mounting has certain applications based on a number of factors. These applications will be discussed in detail for each mounting method.

STUD MOUNTING

Stud mounting is used for permanently mounted transducer applications since it provides the highest frequency response range and is an effective means of keeping the transducer in place. Sometimes an adhesive will be used in combination with stud mounting to prevent the transducer from working its way off of the stud mount. Typically, stud mounting is not practical for collecting PMP route data due to the length of time required to attach and unattach the transducer. However, if very high frequency measurements (> 3,000,000 CPM or 50,000 Hz) are required, stud mounts will have to be used at those data collection points. Also, stud mounting gives extremely repeatable data from measurement to measurement over periods of time.

© Copyright 2001 Technical Associates of Charlotte, P.C.

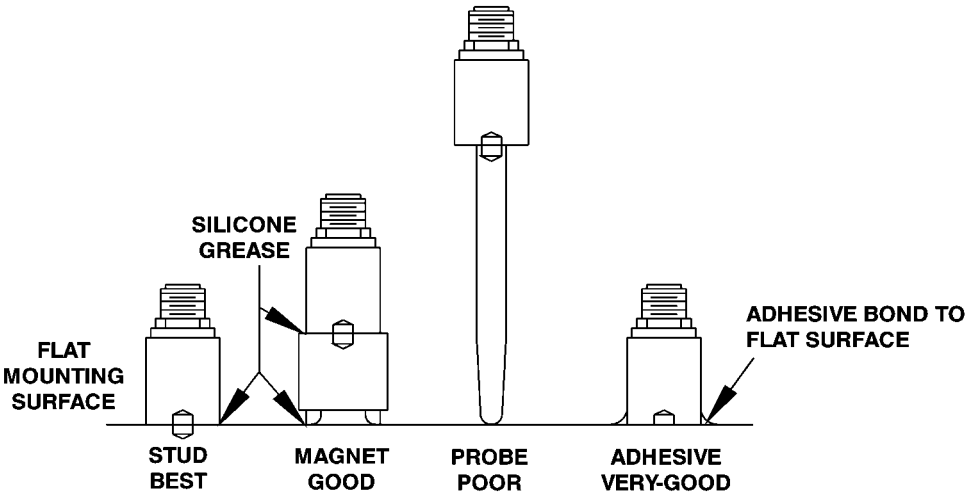


FIGURE 19
MOUNTING METHODS ILLUSTRATED (REF. 1)

ADHESIVE MOUNT

Adhesive mounts also have a very good frequency response range if the proper adhesive type is used and if the proper adhesive thickness is applied (if too thick, this will add damping and degrade the high frequency performance of the mount). However, these mounts may work loose from the machine over long periods of time. Therefore, care should be taken when using an adhesive mount in a permanently mounted transducer configuration. Adhesive mounts are sometimes used in a PMP route when high frequency data is to be collected (>750,000 CPM or 12,500 Hz); however, these are also time consuming. Adhesive mounts are very useful in a diagnostics situation when reliable high frequency data is to be analyzed since a stud mount is rarely found at the point a measurement is desired. Adhesive mounting also gives very repeatable data over a series of measurement surveys.

MAGNETIC MOUNT

Magnetic mounts are the most common used method in PMP programs, as well as when taking diagnostic data. Their response range is generally adequate for most PMP and diagnostic needs (ranging from about 120,000 to 450,000 CPM or 2000 - 7500 Hz). However, it is important to note that some machines such as centrifugal compressors have gear mesh frequencies and harmonics that start at 900,000 CPM (15,000 Hz) and go up to 4,000,000 CPM (almost 70,000 Hz). Data on these machines should be taken with an adhesive or stud mount. A magnetic mount also tends to give reliable repeatability over measurement surveys which is adequate for PMP purposes.

QUICK-CONNECT

Quick-Connect mounts are also ideal for collecting PMP route data since they provide easy mounting and dismounting, as well as allowing a relatively large frequency range that is able to detect most common machinery problems which occur in higher frequencies such as bearing defects. Also, repeatability between measurement surveys is consistent enough for PMP purposes. However, a quick-connect mount should not be used for detecting frequencies above approximately 420,000 CPM (7000 Hz) such as those found in high speed compressor gear mesh frequencies and harmonics.

© Copyright 2001 Technical Associates of Charlotte, P.C.

HANDHELD-PROBE

This is the most undesirable mounting method. The usable frequency range is only up to a maximum of 30,000 to 60,000 CPM (500 - 1000 Hz), no matter the length, diameter or material of the probe! Depending on the length of probe used, the maximum accurate frequency measurable may be as low as 30,000 CPM. With these constraints, the only machinery problems detectable and repeatable will normally be those which occur at 1X, 2X, and 3X RPM for machinery running at 3600 RPM or less. Most importantly, bearing defect frequencies and their harmonics as well as bearing natural frequencies often lie above 30,000 CPM and will be missed entirely if using a hand-held probe. Furthermore, as demonstrated in the research covered in Reference 9, the repeatability of measurement amplitudes and maximum frequency ranges from survey to survey is not consistent and will vary depending on the probe position or amount of hand pressure applied. This impairs the analyst's ability to accurately trend problems and may cause him to miss some problems that occur at the top (or beyond) of the probe's measurable frequency range if insufficient hand pressure was applied during a survey. Handholding a transducer or probe is useful in hard-to-reach places such as where a screen prevents the use of a magnet or other mount type. Also, it is useful if safety is a concern and the analyst cannot safely reach into moving parts of the machine. However, be aware that using this method will miss very important information that is crucial to maintaining a successful PMP program. In the instances described above, it is wiser to attach a permanently mounted transducer in unsafe or hard to reach locations to obtain useful diagnostic data. **Chapter 8 - "COMMON PITFALLS IN EVERYDAY VIBRATION MEASUREMENT"** has an entire section **(8.16 Transducer Mounting and Probes)** devoted to problems that occur with this type of mounting method, as well as real examples of data collected where a bearing is obviously near failure when measured with a magnet or adhesive mount, but does not even show up with a probe type mount. Furthermore, this example shows that probes actually add "phantom" data in the bearing defect range which makes the bearing appear to be in its last stage of failure due to the natural frequency of the probe being excited even though there may be no bearing wear at all.

© Copyright 2001 Technical Associates of Charlotte, P.C.

REFERENCES

1. IRD Mechanalysis; Columbus, OH; Advanced Training Manual, "Vibration Analysis"; 1988.

2. PCB Piezotronics; Depew, NY; Introduction to Piezoelectric Accelerometers, "Fox Valley Modal Workshop"; November, 1990.

3. Lally, R. W.; PCB Piezotronics; Depew, NY; "Transduction"; Tech 689; page 3.

4. Angelo, Martin; Bruel & Kjaer; Naerum, Denmark; "Choosing Accelerometers for Machinery Health Monitoring"; Sound and Vibration Magazine; December, 1990; pages 20-24.

5. IRD Mechanalysis; Columbus, OH; Vibration Technology - 1 Textbook; 1990.

6. Wilcoxon Research; Rockville, MD; Vibration Instrumentation Catalog W-9; July, 1989.

7. Bruel & Kjaer; Naerum, Denmark; Piezoelectric Accelerometers and Vibration Preamplifiers Theory and Application Handbook; March, 1978.

8. Spectral Dynamics (a Scientific-Atlanta Division); San Diego, CA; Vibration Handbook; 1990.

9. Bowers, S. V.; Piety, K. R.; and Piety, R. W.; Computational Systems, Inc.; Knoxville, TN; "Real-World Mounting of Accelerometers for Machinery Monitoring"; Sound and Vibration Magazine; February, 1991; pages 14-23.

10. Computational Systems, Inc. (CSI); Knoxville, TN; "Using the CSI 2110 With Supervisory Systems"; TrendSetter; May, 1991; Vol. 2, No. 3; page 3.

© Copyright 2001 Technical Associates of Charlotte, P.C.

APPENDIX
SPECIFICATIONS FOR VARIOUS TRANSDUCERS FROM A VARIETY OF MANUFACTURERS
(Transducers are sorted alphabetically by vendor name, then numerically for each transducer model number for each vendor)

Sample Transducers are included from each of the following vendors:

- Bently Nevada based in Minden, NV.
- Entek IRD International based in Milford, OH.
- PCB Piezotronics, Inc. based in Depew, NY.
- Rion Co., LTD. based in Tokyo, Japan.
- Vibra Metrics, Inc. based in Hamden, CT.
- Wilcoxon Research based in Rockville, MD.

© Copyright 2001 Technical Associates of Charlotte, P.C.

7200 Proximity Transducer Systems

5 mm and 8 mm

Accurate, reliable measurements

The 7200 5 mm and 8 mm Proximity Transducer Systems are noncontacting, gap-to-voltage transducer systems that measure static as well as dynamic distances between the probe tip and the observed target. The general application is any requirement for an accurate, noncontacting displacement measurement. However, the most common use is for shaft position and vibration measurements on rotating and reciprocating machinery. They are designed for use with virtually any machine type including gas and steam turbines, compressors, pumps, centrifuges, electric motors and generators. The systems offer 80 mils (2 mm) of linear measuring range and are compatible with API 670 requirements.

A system consists of a proximity probe with integral coaxial cable, an extension cable, and Proximitor®. All 7200 proximity transducer systems are compatible with 3300 Monitoring Systems.

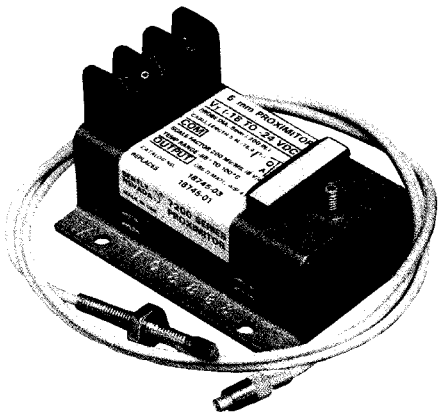
Applications

The 7200 5 and 8 mm Systems have a frequency response of 0 Hz (DC) to 10 kHz and can be used to make the following types of measurements:

- Radial vibration for indicating bearing condition and such machine malfunctions as rotor imbalance and misalignment.
- Axial thrust position for determining thrust bearing wear or potential bearing failure.
- Shaft average radial position, for determining attitude angle, an indicator of rotor stability.
- Vibration amplitude and phase angle for plotting diagnostic information in Polar and Bodé formats.
- Eccentricity to measure the amount of rotor bow.

How a proximity transducer system works

Operating on the eddy current principle, the proximity probe senses the distance between the probe tip and the observed surface. The Proximitor® generates a radio frequency signal, which is radiated through the probe tip into the observed surface. Eddy currents are generated in the surface, and the loss of strength in the return signal is detected by the Proximitor®, which conditions the signal for linear display on a monitor.



7200 Proximity Transducer System

Improved proximity probes

An improved 8 mm version of the 7200 System is now standard which includes Bently Nevada's patented Cable Loc™ feature. This technique of securing the coaxial cable to the probe tip provides the strongest junction yet offered by Bently Nevada and increases the durability of the probe during installation and usage.

Two cable length options are also now available on 8 mm probes. They are 5 metres and 9 metres of integral cable. Since the entire lead-length is built into the probe, an extension cable is not used with these length options.

Nine probe configurations are available to accommodate English and metric thread requirements (See Table 1). The 5 mm probe tips are constructed of fiberglass. The 8 mm probe tips are constructed of polyphenylene sulfide, (PPS), a high performance plastic capable of withstanding unusually harsh, wet and/or chemical environments.

Both transducers are suitable for use in a confined space and offer a scale factor of 200 mV/mil (8 V/mm).

Extension cables

The combination of probe with integral lead and extension cable is designed to achieve a system length of either five or nine metres from probe tip to Proximitor. Probes are available with integral leads of various lengths to match the average extension cable lengths. Probes and cables of the same length, and Proximitors with the same part number, are completely interchangeable.

For added protection, stainless steel armor is available. It provides improved resistance to physical abuse.

Proximitors

The 5 mm and 8 mm systems use the same type of Proximitor®. Both 5 and 9 metre Proximitors are available. A three-conductor, shielded cable is available to provide the signal output and power input between 7200 Proximitors and Bently Nevada monitors. Proximitors can be located up to 1,000 feet (305 metres) from standard Bently Nevada monitors for signal connections.

Hazardous locations

Bently Nevada 7200 Proximity Transducers can be certified by CSA, BASEEFA and FM for use in specific hazardous locations. Refer to Data Sheet L1035 "Machinery Monitoring Systems for Hazardous Locations" for more details.

© Copyright 2001 Technical Associates of Charlotte, P.C.

7200 Proximity Transducer Systems

5 mm and 8 mm

Specifications

Specifications were determined with a –24 Vdc power supply, 10 kΩ load, and an AISI 4140 steel target at +72°F (+22°C).

INPUT

Power: –17.5 Vdc to –26 Vdc at 12 mA maximum consumption.

Leadwire Length: 1,000 feet (305 metres) maximum between Proximitor® and 3300 monitor. Consult manual for frequency roll-off at longer lengths.

OUTPUT

Calibrated Range: 80 mils (2.0 mm). Begins at approximately 10 mils (0.25 mm) from probe face.

Scale Factor: 200 mV/mil (7.87 V/mm), ±4% (measured in increments of 10 mils over the calibrated range) if calibrated as a system. Within ±9.5% including interchangeability errors.

Linearity: Within 0.8 mils (0.02 mm) of a 200 mV/mil straight line if calibrated as a system. Within 2.3 mils (0.06 mm) of a 200 mV/mil straight line including interchangeability errors.

Frequency Response: 0 to 10 kHz (0 to 600,000 cpm); –5% at 10 kHz (600,000 cpm).

Temperature Sensitivity: Typically –3% change in scale factor at +150°F (+65°C) at 50 mils gap (1.27 mm).

Minimum Target Size: .6 inch (15 mm) Diameter.

ENVIRONMENTAL LIMITS

Operating Temperature:

Proximitor®: –60°F to +212°F (–51°C to +100°C).

Probe and Extension Cable: –30°F to +350°F (–34°C to +177°C).

Relative Humidity: Product produced before 01 January 1992: To 95%, non-condensing. Product produced after 01 January 1992: To 100%, condensing, non-submerged (with protection of coaxial connector).

System Weight: 1.3 lbs.(0.59 kg).

Ordering Information

5 mm and 8 mm Standard Mount Probe

A B C D

□□□□□ - □□ - □□ - □□ - □□

Option Descriptions

□□□□□ Probe Part Number Option

Select from Table 1.

A □□ Unthreaded Length Option ①

Order in increments of 0.4 inches [0 4] for English thread, 10 mm [0 1] for metric threads.

English thread configurations:

Maximum unthreaded length:

8.8 inches [8 8]

Minimum unthreaded length:

0.0 inches [0 0]

Example: [0 4] = 0.4 inches

Metric thread configuration:

Maximum unthreaded length:

230 mm [2 3]

Minimum unthreaded length:

0.0 mm [0 0]

Example: [0 6] = 60 mm

B □□ Overall Case Length Option

Order in increments of 0.4 inches [0 4] for English thread, 10 mm [0 1] for metric threads.

English thread configurations:

Maximum case length:

9.6 inches [9 6]

Minimum case length:

0.8 inches [0 8]

Example: [2 4] = 2.4 inches

Metric thread configuration:

Maximum length:

250 mm [2 5]

Minimum length:

20 mm [0 2]

Example: [0 4] = 40 mm

C □□ Total Length Option

05 0.5 metre (20 inches).
Tolerance: +.13 metre, –.05 metre (+5 inches, –2 inches).

10 1.0 metre (39 inches).
Tolerance: +.25 metre, –.10 metre (+10 inches, –4 inches).

50 5 metre (195 inches).③
Tolerance: +.6 metre, –.5 metre (+24 inches, –20 inches).

90 9 metre (351 inches).
Tolerance: +1.0 metre, –.9 metre (+39 inches, –35 inches)

D □□ Connector Option

00 Without connector.

02 With miniature male coaxial connector.

8 mm Reverse Mount Probe

A B C D

21508 - 02 - 12 - □□ - □□

Option Descriptions

C □□ Total Cable Length Option

05 0.5 metre (20 inches).
Tolerance: +.13 metre, –.05 metre (+5 inches, –2 inches).

10 1.0 metre (39 inches).
Tolerance: +.25 metre, –.10 metre (+10 inches, –4 inches).

50 5 metre (195 inches).③
Tolerance: +.6 metre, –.5 metre (+24 inches, –20 inches).

90 9 metre (351 inches).
Tolerance: +1.0 metre, –.9 metre (+39 inches, –35 inches)

D □□ Connector Option

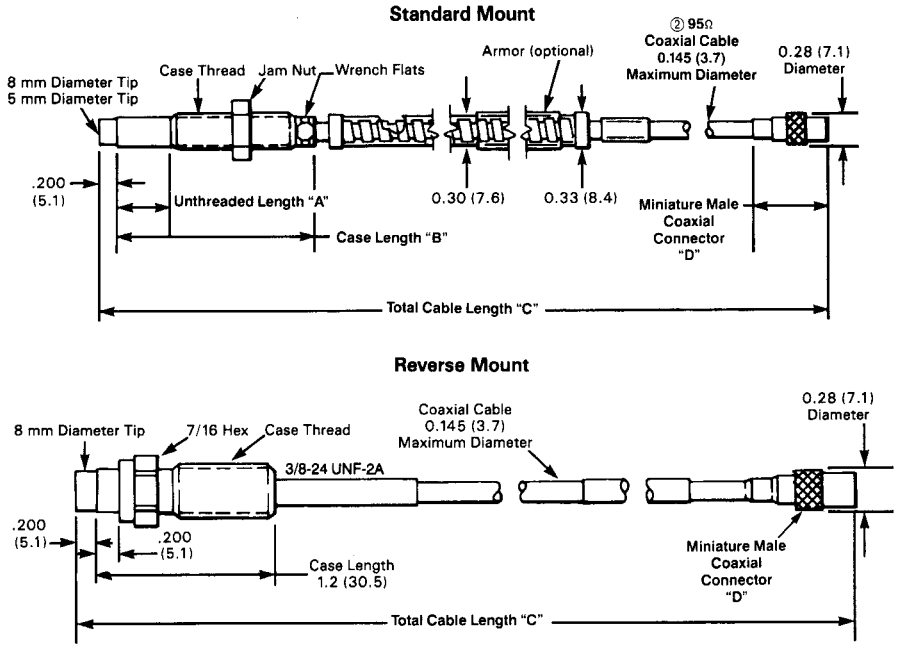
00 Without connector.

02 With miniature male coaxial connector.

© Copyright 2001 Technical Associates of Charlotte, P.C.

7200 Proximity Transducer Systems

5 mm and 8 mm



5 mm and 8 mm Proximity Probe Configurations ④ ⑤

TABLE 1
Probe Part Number Option Descriptions

TIP SIZE		CASE		
PART NO. □□□□	DIAMETER	CONFIGURATION TYPE	THREADS	STAINLESS STEEL LEAD ARMOR
*21500-	5 mm	Standard	1/4-28	No
*21501-	5 mm	Standard	1/4-28	Yes
21504-	8 mm	Standard	3/8-24	No
21505-	8 mm	Standard	3/8-24	Yes
21508-02-12	8 mm	Reverse	3/8-24	No
22810-	8 mm	Standard	M10 × 1	Yes
22811-	8 mm	Standard	M10 × 1	No
*22812-	5 mm	Standard	M8 × 1	Yes
*22813-	5 mm	Standard	M8 × 1	No

**** Maximum case length for all M8 × 1 probes is 100 mm; and for 1/4-28 probes is 4 inches. Also, maximum overall length for 5 mm probes is 1 metre.***

NOTES:

① *Unthreaded length must be at least 0.8 inches (20 mm) less than case length.*

② *Maximum diameter for 21500 and 21501 probes is 0.111 in. (3.4 mm).*

③ *For use with the 18745-03 five-metre Proximitor® only.*

④ *Letters inside quotation marks refer to probe ordering options.*

⑤ *Dimensions in parentheses are in millimetres.*

© Copyright 2001 Technical Associates of Charlotte, P.C.

7200 Proximity Transducer Systems

5 mm and 8 mm

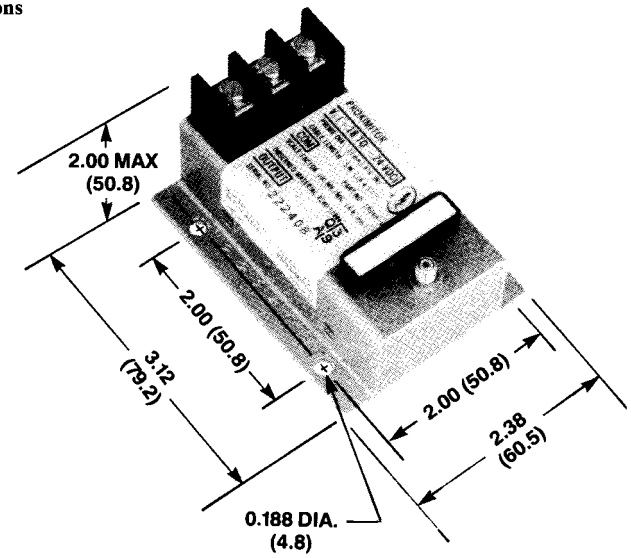
Proximitor®

18745-□□

03 For combined probe and extension cable system length of 5 metres (16.4 feet).

04 For combined probe and extension cable system length of 9 metres (29.6 feet).

Dimensions



Dimensions in parentheses are in millimetres.

Extension Cable

A B
21747 - □□□ - □□

Option Description

A □□□ Cable Length Option ⑥

040 4.00 metres (157 inches).⑦

045 4.50 metres (177 inches).⑦

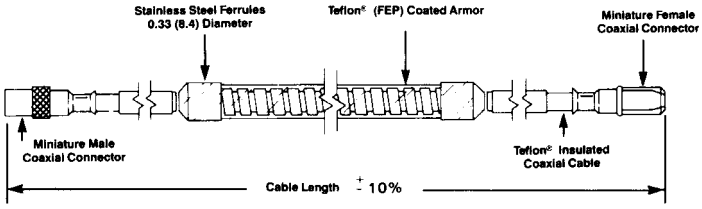
080 8.00 metres (315 inches).

085 8.50 metres (335 inches).

B □□ Armor Option

00 Without armor.

01 With armor.



NOTES:

⑥ *Extension cable physical length equals the electrical length, ± 10%*

⑦ *For use with the 18745-03 five-metre Proximitor® only.*

© Copyright 2001 Technical Associates of Charlotte, P.C.

Velocity Transducer Systems

86205

Specifications

Specifications are at +72°F (+22°C) and 1 in/sec (25 mm/s) at 100 Hz (6,000 cpm) unless otherwise specified.

ELECTRICAL

Frequency Response: 10 Hz to 1000 Hz (600 cpm to 60,000 cpm), –3dB typical.

Sensitivity: 500 mV/in/sec (20 mV/mm/s) zero-to-peak ±5% at 100 Hz (6,000 cpm) and 1 in/sec (25 mm/s) with a mounting tolerance of ± 2° for both vertical and horizontal orientations when operated with a 24.9 kΩ load.

Orientation Sensitivity Change: ≤2% at ±20° (vertical) or ±10% (horizontal).

Input Load: 24.9 kΩ ±1%.

Amplitude Linearity Error: ±5% at 100 Hz (6,000 cpm) from 0.1 to 4.0 in/sec (2.5 to 100 mm/s) zero-to-peak.

Amplitude Range: 32 mils (0.81 mm) peak-to-peak, maximum.

Transverse Sensitivity: <12% typical over operating frequency.

Coil Resistance: 395 Ω ±5%.

Temperature Coefficient: -0.04%/°F (.022%/°C).

Polarity of Output Signal: When case movement is toward the connector, the center male pin provides positive signal relative to the adjacent pins.

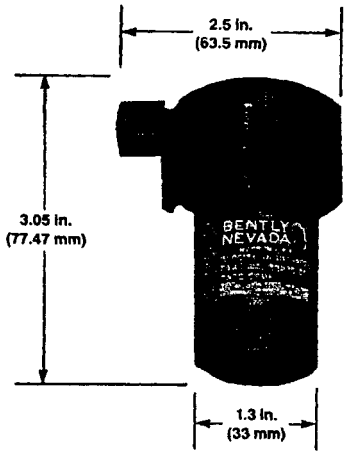
ENVIRONMENTAL LIMITS

Temperature Range:

Operating: –22°F to +180°F (–30°C to +82°C).

Storage: –30°F to +180°F (–34°C to +82°C).

Relative Humidity: 100%, nonsubmerged. (NEMA 4X requirement).



86205 Velocity Transducer Dimensions

PHYSICAL

Reference Height: 3.05 inches (77 mm) with cap assembly.

Case Body Diameter:

Cap: 1.8 inches (46 mm) nominal.

Body: 1.3 inches (33 mm) nominal.

Reference Weight: 7 ounces (200g)

Case Material: Molded polyphenylene sulfide with 304 SST threaded insert.

Mounting: 3/8-24 UNF-2B female threaded insert is standard. Adhesive-mount base kits and threaded mounting studs are available by ordering the standard Mounting Adapter option.

MECHANICAL

Maximum Shock: 50g (490 m/s²).

Ordering Information

Velocity Seismoprobe®

A 5-pin cable connector and cover are included for cable connection.

	A	B
86205 -	□□	**-** □□

Option Descriptions

A □□ Transducer Orientation Option

01 Vertical (0° ± 20°)

02 Horizontal (90° ± 10°)

B □□ Mounting Adapter Option

01 3/8-24 UNF

02 1/2-20 UNF

03 Adhesive kit, 3/8-24 UNF

04 M6 × 1

05 Adhesive kit, M6 × 1

06 1/4-28 UNF

ACCESSORIES

Transducer Cable

4-conductor cable with drain wire; 18 AWG (1.0 mm²); ordered in rolls (refer to ordering information below). Cable specifications are available in the Trendmaster® 2000 data sheet L4110. Maximum recommended cable length per transducer is 98 feet (30 metres).

	A	
85033 -	□□	**- 00**

Option Descriptions

A □□ Cable Roll Length Option

01 500 ft (150 metres)

02 1000 ft (305 metres)

Cable Connectors

When ordering the transducer, a connector to the transducer cable is provided. The following spare parts are available.

00523103	5-pin, 18 AWG (1.0 mm²) connector for interfacing cable to transducer.
00523104	Cover for 00523103 connector.

© Copyright 2001 Technical Associates of Charlotte, P.C.

Velocity Transducer Systems

86205

Specifications

Specifications are at +72°F (+22°C) and 1 in/sec (25 mm/s) at 100 Hz (6,000 cpm) unless otherwise specified.

ELECTRICAL

Frequency Response: 10 Hz to 1000 Hz (600 cpm to 60,000 cpm), –3dB typical.

Sensitivity: 500 mV/in/sec (20 mV/mm/s) zero-to-peak ±5% at 100 Hz (6,000 cpm) and 1 in/sec (25 mm/s) with a mounting tolerance of ± 2° for both vertical and horizontal orientations when operated with a 24.9 kΩ load.

Orientation Sensitivity Change: ≤2% at ±20° (vertical) or ± 10% (horizontal).

Input Load: 24.9 kΩ ±1%.

Amplitude Linearity Error: ±5% at 100 Hz (6,000 cpm) from 0.1 to 4.0 in/sec (2.5 to 100 mm/s) zero-to-peak.

Amplitude Range: 32 mils (0.81 mm) peak-to-peak, maximum.

Transverse Sensitivity: <12% typical over operating frequency.

Coil Resistance: 395 Ω ±5%.

Temperature Coefficient: -0.04%/°F (.022%/°C).

Polarity of Output Signal: When case movement is toward the connector, the center male pin provides positive signal relative to the adjacent pins.

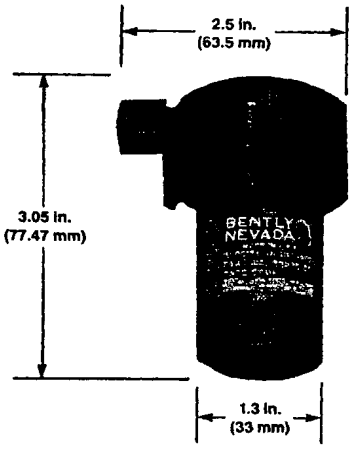
ENVIRONMENTAL LIMITS

Temperature Range:

Operating: –22°F to +180°F (–30°C to +82°C).

Storage: –30°F to +180°F (–34°C to +82°C).

Relative Humidity: 100%, nonsubmerged. (NEMA 4X requirement).



86205 Velocity Transducer Dimensions

PHYSICAL

Reference Height: 3.05 inches (77 mm) with cap assembly.

Case Body Diameter:

Cap: 1.8 inches (46 mm) nominal.

Body: 1.3 inches (33 mm) nominal.

Reference Weight: 7 ounces (200g)

Case Material: Molded polyphenylene sulfide with 304 SST threaded insert.

Mounting: 3/8-24 UNF-2B female threaded insert is standard. Adhesive-mount base kits and threaded mounting studs are available by ordering the standard Mounting Adapter option.

MECHANICAL

Maximum Shock: 50g (490 m/s²).

Ordering Information

Velocity Seismoprobe®

A 5-pin cable connector and cover are included for cable connection.

	A	B
86205 -	□□	**-** □□

Option Descriptions

A □□ Transducer Orientation Option

01 Vertical (0° ± 20°)

02 Horizontal (90° ± 10°)

B □□ Mounting Adapter Option

01 3/8-24 UNF

02 1/2-20 UNF

03 Adhesive kit, 3/8-24 UNF

04 M6 × 1

05 Adhesive kit, M6 × 1

06 1/4-28 UNF

ACCESSORIES

Transducer Cable

4-conductor cable with drain wire; 18 AWG (1.0 mm²); ordered in rolls (refer to ordering information below). Cable specifications are available in the Trendmaster® 2000 data sheet L4110. Maximum recommended cable length per transducer is 98 feet (30 metres).

	A	
85033 -	□□	**- 00**

Option Descriptions

A □□ Cable Roll Length Option

01 500 ft (150 metres)

02 1000 ft (305 metres)

Cable Connectors

When ordering the transducer, a connector to the transducer cable is provided. The following spare parts are available.

00523103	5-pin, 18 AWG (1.0 mm²) connector for interfacing cable to transducer.
00523104	Cover for 00523103 connector.

© Copyright 2001 Technical Associates of Charlotte, P.C.

Velocity Transducer Systems

86205

Cable Connection Tools

85718-01 Crimp tool assembly for attaching connectors to cable. One is required for cable installation.

89138-01 Cable armor adaptor; 5/16″ armor.

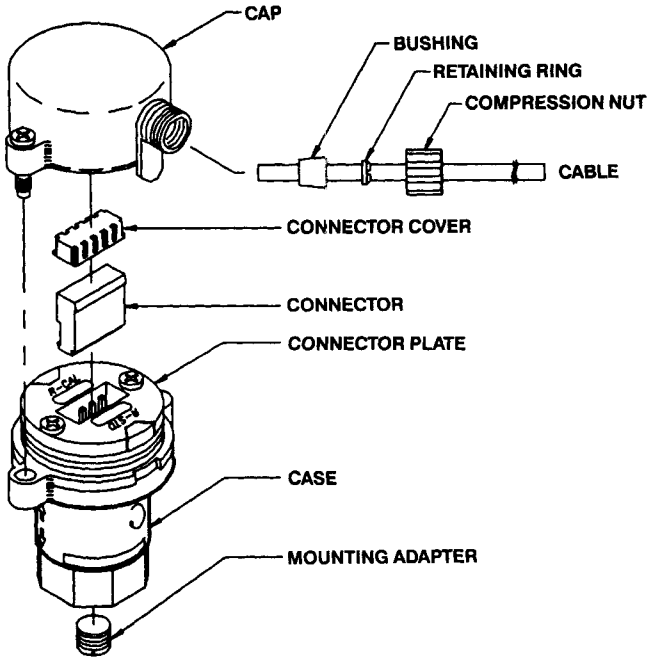
Adhesive-mount Base Kits

Adhesive-mount base kits are available for quick and easy transducer mounting. The kit contains materials (adhesive and bases) for adhesive-mount bases for Velocity Seismoprobes®.

⚠CAUTION

If casing vibration measurements are being made for the overall protection of a machine, thought should be given to the usefulness of the measurement for each application. Most common machine malfunctions, such as unbalance, misalignment, etc., occur on the rotor and originate as an increase (or at least a change) in rotor vibration. For any casing measurement alone to be effective for overall machine protection, a significant amount of rotor vibration must be faithfully transmitted to the machine casing or mounting location of the transducer.

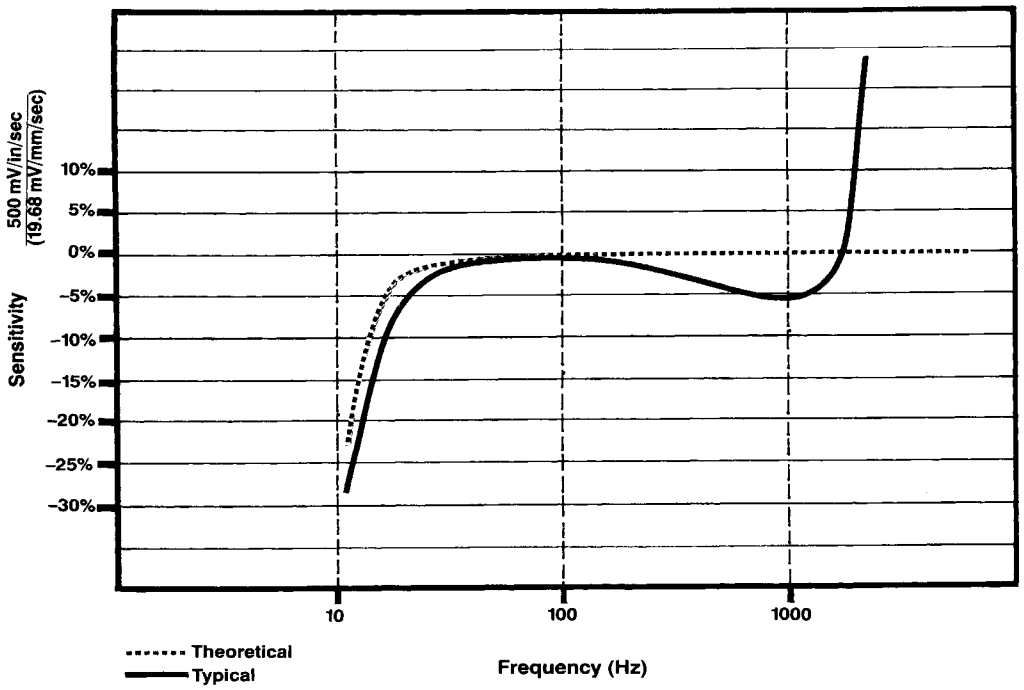
In addition, care should be exercised in the physical location of the velocity sensor on the bearing housing or machine casing. Improper installation may result in a decrease of the transducer amplitude and frequency response and/or the generation of false signals that do not represent vibration on that particular machine. For more information, refer to the appropriate Instruction Manuals and Application Notes.



© Copyright 2001 Technical Associates of Charlotte, P.C.

Velocity Transducer Systems

86205



86205 Velocity Transducer Frequency Response

© Copyright 2001 Technical Associates of Charlotte, P.C.

Dual Probe Transducer System

26530

Measures absolute shaft motion on turbine generators

Large rotating machinery can often have substantial absolute shaft motion, which is the result of rotor vibration combined with excessive bearing housing vibration. Absolute vibration consists of radial vibration of the rotor **and** seismic vibration of the bearing housing or machine case. When significant bearing housing vibration exists, a Dual Probe Transducer System can provide additional information on the condition of the machine. Bently Nevada recommends the Dual Probe System when **more than one-third** of the shaft vibration is transmitted to the bearing housing.

The Dual Probe's primary function is to measure absolute motion on a machine. From this measurement, you're able to obtain vital diagnostic information on rotor position, vibration wave form, and phase angle.

Two types of measurements are required to determine absolute rotor motion: casing or foundation vibration and rotor motion relative to the bearing. The Dual Probe uses an eddy current proximity probe to measure rotor motion and a Velocity Seismoprobe® transducer to measure casing or foundation vibration.

A Dual Probe Monitor vectorially sums the two signals to yield an absolute motion measurement. Prior to summation, the proximity and integrated velocity signals are available at the monitor for examination.

The Dual Probe Transducer System consists of a proximity probe, a Velocity Seismoprobe® transducer, adapter, side fittings, and a probe support/oil seal. It is designed to be weatherproof when properly installed. The Dual Probe is used in conjunction with the 3300/65 Dual Probe Monitor.

Types of measurements

By using the Dual Probe to observe absolute motion during fixed running speed and startup and shutdown, new insights can be found on the machine's mechanical condition. You can measure:

- Rotor motion relative to bearing motion.

Dual Probe Transducer System

- Rotor absolute motion relative to free space.
- Rotor position relative to the bearing.
- Bearing cap velocity.

Many machinery vibration problems begin with rotor position changes within the bearing clearance. By making rotor position and rotor motion measurements, it's possible to identify a fault before it becomes evident from other measurement parameters.

For complete machinery monitoring diagnostics using orbit analysis and shaft centerline position information, two Dual Probe assemblies in an XY configuration are required per radial bearing. Thus, Dual Probe assemblies are often used in pairs on each bearing to allow Polar, Orbit and Timebase formats used in balancing and machine analysis. However, they can be used as a standard single channel absolute vibration measurement in a wide variety of applications.

Additional information on the Dual Probe is available in an Applications Note entitled **Bently Nevada Dual Probe Versus Shaft Rider** (AN006). Also available are reprints: **Evaluating TSI Instrumentation for Retrofit Program** (L8105) and **An Expanding Role for Turbine Supervisory Instrumentation** (L8108).

Proximity probe

The Dual Probe Proximity Transducer is available in high performance plastic capable of withstanding unusually harsh, wet and/or chemical environments.

The probe coil diameter of both versions is 8 mm. The probes are offered in one-metre and half-metre lead lengths.

Other Bently Nevada Proximity Probes are available upon request. For more information, please contact your nearest Bently Nevada sales representative.

Velocity Seismoprobe® Transducer

The Velocity Seismoprobe® Transducer is available in standard and high temperature versions. It is ordered according to the intended mounting angle.

Adapter

The standoff adapter provides clearance between the probe housing and machine casing. It is ordered according to the length required and is available in 7/8-14 UNF, 1-11½ NPT, and 3/4-14 NPT mounting threads.

Side fittings

Side fittings are used to connect conduit or cabling to the Dual Probe housing. Four configurations are available.

Probe Support/Oil Seal

The Probe Support/Oil Seal is available as an option. It can protect the transducer system from oil and provides additional support for the probe sleeve.

Some type of sleeve support is required for sleeve lengths greater than 12 inches (305 mm). Without additional support, sleeve resonances may develop, causing inaccurate relative probe measurements or sleeve failure.

© Copyright 2001 Technical Associates of Charlotte, P.C.

Dual Probe Transducer System

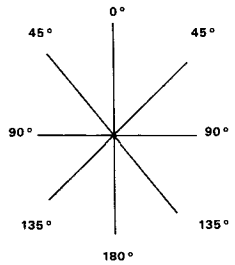
26530

Specifications

Relative Probe Adjustment

Adapter Mounting Thread	Dimension A on Drawing	Adjustment Range From Nominal Setting of Dimension B on Drawing	
		Toward Target	Away From Target
Without Adapter	Not Applicable	1.02 inches (25.9 mm)	1.03 inches (26.2 mm)
7/8-14 UNF	Less than 2 inches (51 mm)	1.02 inches (25.9 mm)	1.03 inches (26.2 mm)
1-11½ NPT	Less than 2 inches (51 mm)	0.85 inch (21.6 mm)	0.86 inch (21.8 mm)
3/4-14 NPT	Less than 2 inches (51 mm)	0.88 inch (22.4 mm)	0.89 inch (22.6 mm)

Seismoprobe® Orientation



Seismoprobe® angular range from 0° vertical as viewed from rotor end.

Ordering Information

A B C D E
26530 - □□ - □□ - □□ - 0□□ - □□□
F G H
- □□ - □□ - □□

Option Descriptions

A □□ Proximity Probe Option

00 Probe not required
05 7200 Series 8 mm probe
12 3300 Series 8 mm probe
19 3300 Series 5 mm probe.

B □□ Probe Lead Length Option

00 Probe not required
05 0.5 metre
10 1.0 metre

C □□ Adapter Thread Option

00 Adapter not required
01 7/8-14 UNF
02 1-11½ NPT
03 3/4-14 NPT

D 0□□ Adapter Length Option
(Dimension A on dimension drawing)
Order in increments of 0.1 inch (2.5 mm).
Minimum length: 1.7 inches
0 1 7 (43 mm)
Maximum length: 2.0 inches
0 2 0 (51 mm)
If you do not require an adapter, please enter 0 0 0.

E □□□ Probe Penetration Option
(Dimension B on dimension drawing)
Order in increments of 0.1 inch (2.5 mm).
Minimum length: 2.0 inches
0 2 0 (51 mm)
Maximum length is dependent upon the Adapter Length (Dimension A).
For 0.5 metre probe cable:
Maximum length of Dimension A + Dimension B is
13.3 inches 1 3 3 (338 mm).
For 1.0 metre probe cable:
Maximum length of Dimension A + Dimension B is
31.0 inches 3 1 0 (787 mm).

F □□ Velocity Seismoprobe® ① Transducer Option

00 Without Velocity Seismoprobe® Transducer

For 600 cpm Minimum Operating Frequency:

Seismoprobe® Type		Angular Range
21	Standard, 2 wire	0° ± 100°
22	High temperature, 2 wire,	0° ± 100°

G □□ Fittings Option

00 Not required
01 One 3/4 NPT conduit fitting and one 3/4 NPT plug
02 Two 3/4 NPT conduit fittings
03 One 3/4 NPT plug, one 3/4 to 1/2 NPT reducer, and one cable seal grip with grommet
04 One 3/4 NPT plug

H □□ Probe Support/Oil Seal Option

00 Not required
01 Required

Notes:
① *Standard temperature Seismoprobe® rated –20°F to +250°F (–29°C to +121°C). High temperature Seismoprobe® rated –20°F to +400°F (–20°C to +204°C).*

© Copyright 2001 Technical Associates of Charlotte, P.C.

Dual Probe Transducer System

26530

ACCESSORIES

40467 - [0][ ][0] - [ ][ ][0] Individual Sleeve (A, B)

Option Descriptions

To order an individual sleeve, add Adapter Length (Dimension A) and Probe Penetration (Dimension B) plus 2.4 inches (61 mm) for length inside housing.

Order in increments of 0.1 inches

Maximum Dimension A: 2.0 inches [0][2][0][0] (51 mm)

Minimum Total Sleeve Length: 4.4 inches [0][4][4][0] (112 mm)

Maximum Total Sleeve Length: 33.0 inches [3][3][0][0] (838 mm)

The recommended maximum Total Sleeve Length is:

For 0.5 metre probe cable: 13.3 inches [1][3][3][0] (338 mm)

For 1.0 metre probe cable: 31.0 inches [3][1][0][0] (787 mm)

40468 - [ ][ ] - [0][ ][0] Individual Standoff Adapter (A, B)

Option Descriptions

A [ ][ ] Base Thread Option

01 7/8 - 14 UNF

02 1 - 11 1/2 NPT

03 3/4 - 14 NPT

B [0][ ][0] Length Option

Order in increments of 0.1 inches.

Minimum Length for Base Thread Option:

–01: 1.70 inches [0][1][7][0]

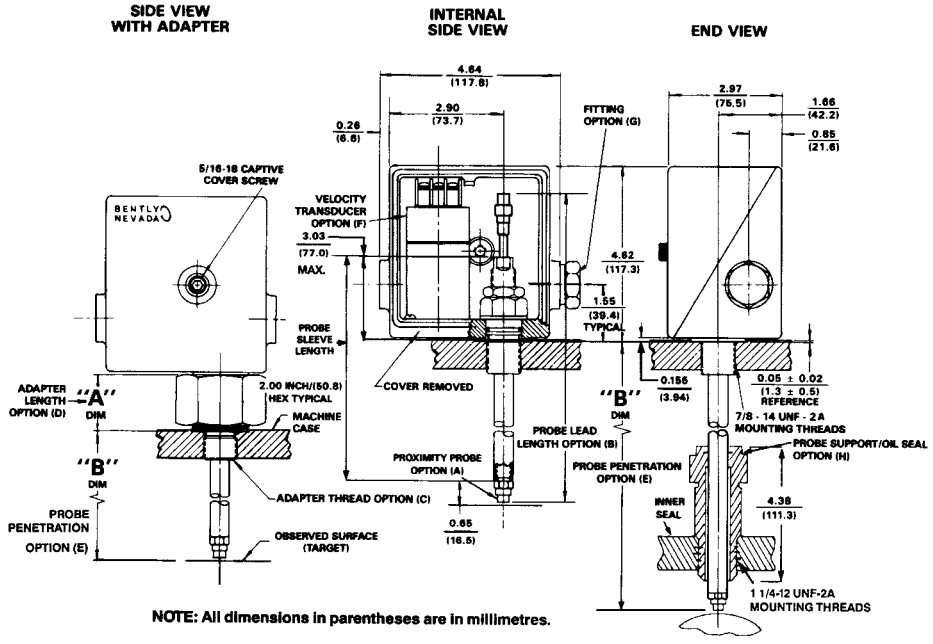
–02: 1.90 inches [0][1][9][0]

–03: 1.80 inches [0][1][8][0]

Maximum length: 2.0 inches (51 mm) [0][2][0][0]

37948-01 Individual Probe Support/Oil Seal made from Ertalon.

TW8029326 Instruction Manual



NOTE: All dimensions in parentheses are in millimetres.

© Copyright 2001 Technical Associates of Charlotte, P.C.

Model 911

Accelerometer **EFFECTIVE: Preliminary**

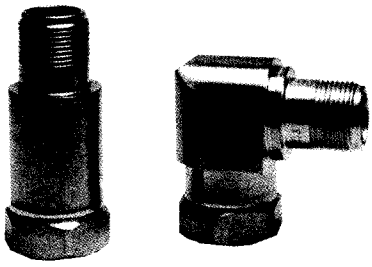
DESCRIPTION

The IRD 911 is a solid-state seismic accelerometer for permanent mounting on machinery. It is manufactured in a fully welded 316L SSTL housing with a hermetic sealed connector, that will withstand harsh industrial environments including high temperatures and splashing fluids.

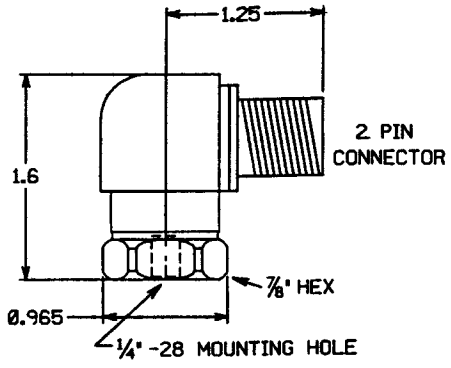
The IRD 911's wide frequency and amplitude ranges accommodate most types of rotating machines. The small size permits installation near bearings.

The IRD 911 has an internal, low-output-impedance amplifier capable of transmitting a vibration signal accurately through several thousand feet of cable, ideal for large installations with centralized signal processing equipment. A D.C. bias voltage on the output facilitates remote testing for cable continuity and proper operation.

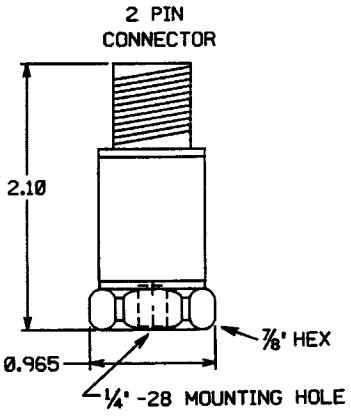
The IRD 911 can be mounted directly on the machine, via a 1/4 28 screw. The IRD 911 accelerometer is available in a straight or a right angle mounted connector configuration.



OUTLINE DIMENSIONS



Model 911R



Model 911S

ENTEK IRD INTERNATIONAL

© Copyright 2001 Technical Associates of Charlotte, P.C.

Specifications for	911 Accelerometer	Units
Sensitivity, nom	100	mV/g
Frequency Response ± 10% - 3dB	 1.5 - 7k 1	 Hz Hz
Dynamic Range	001 - 75	g peak
Resonant Frequency, mounted, nom	22	kHz
Transverse Sensitivity, avg	5	%
Temperature Range	-40 to + 300 -40 to + 148	°F °C
Temperature Coefficient	.025	% / °F
Magnetic Field Susceptibility	100	µg/Gauss
Base Strain Sensitivity	.005 max	g/µstrain
Construction (case and armor)	316L-series stainless steel	
Seal	Hermetic	
Environmental Protection	Meets NEMA 4x, 12, 13, IEC IP66	
Grounding	Case isolated from circuit	
Power	+ 18 to + 26 VDC source through 1 to 4 mA Constant - current Diode	
Output Impedance	100	ohms
Output Bias	+ 10 to + 18	VDC

ORDERING INFORMATION

(Qty) P/N02402 - PVC Twisted Pair Shielded
(Qty) P/N 04366 - Mating Cable Connector
(Qty) P/N37691 - Model 911S Accelerometer (Straight Connector)
(Qty) P/N37692 - Model 911R Accelerometer (Right Angle Connector)
(Qty) P/N30539 - Tefezel Twisted Pair Shielded
(Qty) P/N 04636 - Rubber Insulated Shielded Pair

ENTEK IRD

www.entekird.com

Corporate Headquarters
1700 Edison Drive
Milford, OH
USA 45150
Tel: (1) 513-576-6151
Fax: (1) 513-576-6104

Australia Operations
Suite 502
655 Pacific Highway
St. Leonards NSW
Australia 2065
Tel: (61) 2 9436 4422
Fax: (61) 2 9436 4355

China Operations
Ceroil Plaza
Room 901/902
9 Yide Street,
Zhongshan Dist.
Dalian 116001, China
Tel: (86) 411 265 6594
Fax: (86) 411 265 6627

European Operations
Bumpers Lane
Sealand Industrial Estates
Chester
UK CHI 4LT
Tel: (44) 1244 374914
Fax: (44) 1244 379870

India Operations
Plot 1/5 Marol Co-op. Ind. Est.
Andheri (East)
Mumbai
400 059
Tel: (91-22) 852 2906
Fax: (91-22) 852 1814

FORM # 5202

© Copyright 2001 Technical Associates of Charlotte, P.C.

Model 943

Accelerometer and Accessories

EFFECTIVE: December 1992

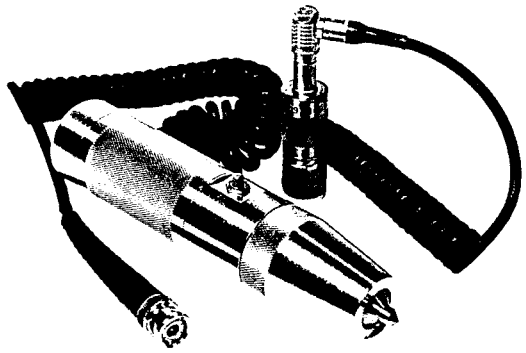
DESCRIPTION

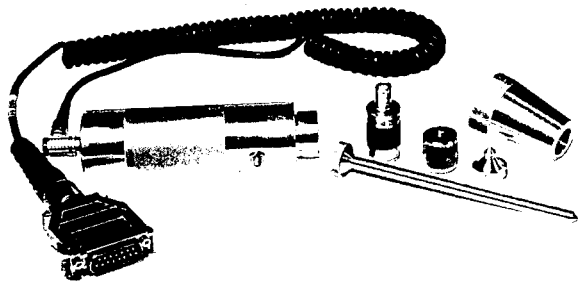
The IRD 943 is a solid-state seismic accelerometer specifically designed for use with portable analyzers or data collectors to measure vibration levels on mechanical equipment. The 943 design is based on the proven performance of the 942 transducer used in permanent applications, and it exceeds the performance of the 970 transducer for portable applications. The primary advantage of the 943 are its small and light weight packaging and the applications flexibility.

A complete kit of accessories is available for the 943, including a small rare-earth magnet, 2 probe options, and a handle with built-in STORE button. The 943 can be stud-mounted or used with a magnet or probe in collecting data. The remote STORE function speeds up the data collection process by providing "hands-free" operation of the data collector and allows complete concentration on the measurement itself.

The 943 incorporates the industry standard ICP interface circuit and has a sensitivity of 100 mv/g. For walk-around data collection routes combining permanently installed 942's and hand held measurements, the 943 provides a distinct advantage where there is no need for switching cables, input ports or transducer sensitivity.

The 943 incorporates a low-output-impedance, constant-current driven amplifier with low noise floor and utmost accuracy across the entire frequency range. The wide frequency and amplitude ranges of the 943 accommodate most vibration measurement applications for predictive maintenance, including frequencies in the range of 120 CPM to 720,000 CPM and ultrasonic Spike Energy™ signals. The small size of the 943 permits its use in remote and tight spaces. All 943 components are designed for the highest reliability in withstanding harsh industrial environments, including high temperatures and splashing fluids.





The 943 transducer and accessories are ideally suited for the IRD FAST TRACK, 838, 885, and 890 portable instruments. The following cable compatibility matrix is provided for ordering convenience.

	INSTRUMENT			
Cable Part Numbers	FAST TRACK	838*	885	890
w/o remote STORE	36365	36365	36365	36365
with remote STORE	35976	-	36733	36733

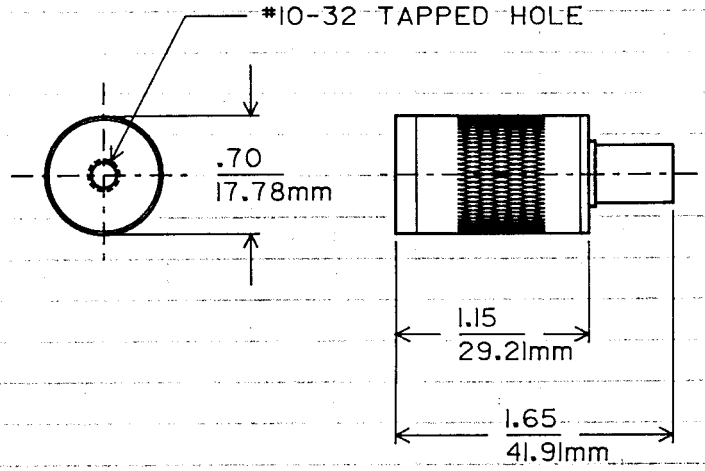
*Note: The remote STORE function is not supported by the 838 instrument

ENTEK IRD INTERNATIONAL

© Copyright 2001 Technical Associates of Charlotte, P.C.

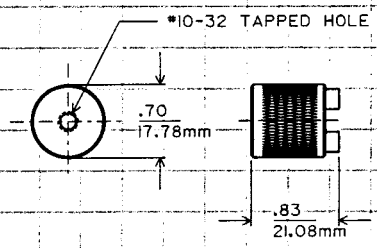
OUTLINE DIMENSIONS

943 Accelerometer P/N 35530

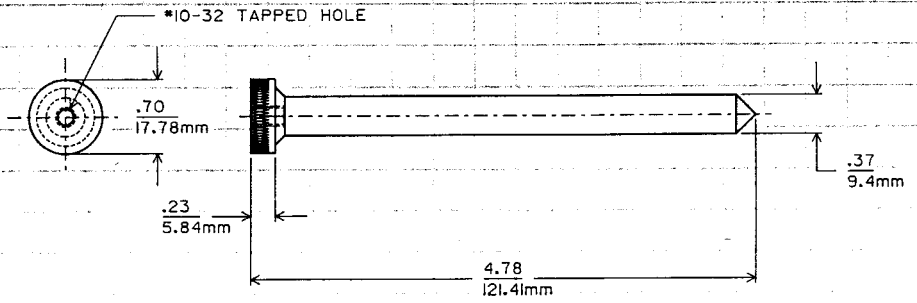


The 943 accelerometer can be used with either a magnetic holder (P/N 35536) or 5 inch probe (P/N 35540) as shown below.

Magnetic Holder P/N 35536



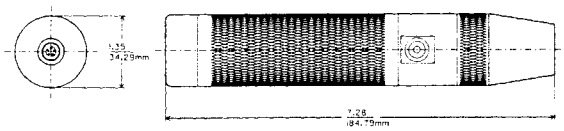
5 inch probe P/N 35540



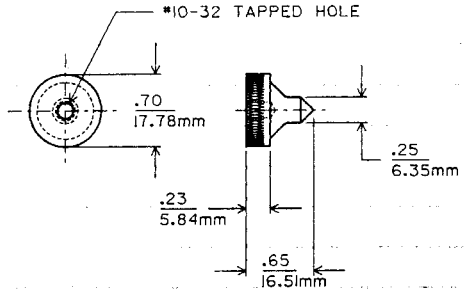
© Copyright 2001 Technical Associates of Charlotte, P.C.

The 943 can also be used in combination with handle (P/N 35541) and choice of 5 inch probe or 1/2 inch probe P/N 35973) as shown below.

943 Handle with STORE button P/N 35541



1/2 inch probe P/N 35973

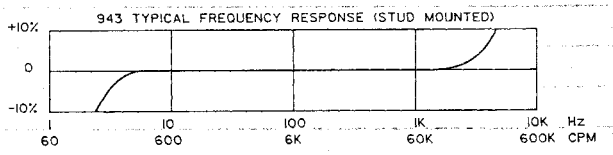


Assembly Instructions for the Use of the Handle

The handle nose cone is removed, the 943 accelerometer is snapped into place, either probe is attached to the 943, and the nose cone is re-installed to lock all components together. The transducer cable is then attached to the end of the handle. The integral STORE button in the handle operates with compatible data collectors (such as the IRD FAST TRACK).

Specifications for	943 Accelerometer	Units
Sensitivity, nom	100	mV/g
Acceleration range	0-50	g peak
Resonant Frequency, stud mounted, nominal	25 k	Hz
Frequency Response, stud mounted	3.5-7 k	Hz ± 10%
Frequency Response, stud mounted	2-12 k	Hz ± 3 dB
Temperature Range	-60 to + 250 -52 to + 121	°F °C
Transverse Response, avg	5	%
Magnetic Field Sensitivity	100	µg/gauss
Base Strain Sensitivity, stud mounted	.01	g/µ strain
Shock Limit	1000	g peak
Output Impedance	100	ohm
Seal	Epoxy	
Case Material	304 Stainless Steel	
Connector	Gold plated contacts; chrome plated brass shell	
Power Requirement	1-4 mA constant current; + 16 to + 24 VDC	
Grounding	Case connected to circuit common	
Weight (including connector)	1.55 44	oz grams
Environmental Protection	Waterproof, dustproof, corrosion-resistant per NEMA 4X; Submersible to 50 psi per NEMA 6 and IEC IP67 (100 hours)	

943 Typical Frequency Response (stud mounted)



© Copyright 2001 Technical Associates of Charlotte, P.C.

MAGNET P/N 35536 SPECIFICATIONS

Frequency @ + 3dB response 5000 Hz (300 k CPM)		
Case Material	303 Stainless Steel	
Dimensions	Diameter	.70 in (17.8 mm)
	Length	.75 in (19.5 mm)
	Weight	1.06 oz (30 g)

PROBE P/N 35973 SPECIFICATIONS

Frequency @ + 3dB response 1400 Hz (84 k CPM)		
Material	303 Stainless Steel	
Dimensions	Diameter (Shaft)	.25 in (6.35 mm)
	Length	.65 in (16,5 mm)
	Weight	.53 oz (15 g)

PROBE P/N 35540 SPECIFICATIONS

Frequency @ + 3dB response 1000 Hz (60 k CPM)		
Material	303 Stainless Steel	
Dimensions	Diameter (Shaft)	.36 in (0.1 mm)
	Length	4.77 in (121.3 mm)
	Weight	1.48 oz (42 g)

HANDLE P/N 35541 SPECIFICATIONS

Switch	Momentary Push Switch, Hermetically Sealed	
Material (structural parts)	Aluminum	
Dimensions	Diameter	1.35 in (34.3 mm)
	Length	6.87 in (175 mm)
	Weight	6.07 oz (172 g)

CABLE P/N 35976 SPECIFICATIONS

Connectors	Right Angle 943; Submin-D FAST TRACK Data Collector
Cable	Retractile; Neoprene Jacket; 72 in (1829 mm) extended

CABLE P/N 36365 SPECIFICATIONS

Connectors	Right Angle 943; BNC
Cable	Retractile; Neoprene Jacket; 72 in (1829 mm) extended

ORDERING INFORMATION

P/N 36394 - Complete Kit for IRD FAST TRACK includes: Accelerometer, Cable (P/N 35976), 2 probes, magnet & handle with built-in store button

Individual components can be ordered from the chart below:

P/N 35530 - Model 943 Accelerometer
P/N 35976 - 943 Cable to FAST TRACK (w/ remote STORE), 18 in (45.7cm) to 6ft (1.83m) **
P/N 36365 - 943 BNC instrument cable (no remote STORE), 18 in (45.7 cm) to 6 ft (1.83M)
P/N 36733 - 943 Cable to 885, 890 (w/ remote STORE), 18 in (45.7 cm) to 6 ft (1.83M)
P/N 35973 - 943 Probe 1/2 in (1.3cm)
P/N 35540 - 943 Probe 5 in (12.7 cm)
P/N 35536 - 943 Magnetic holder
P/N 35541 - 943 Handle with STORE button

ENTEK IRD

www.entekird.com

Corporate Headquarters
1700 Edison Drive
Milford, OH
USA 45150
Tel: (1) 513-576-6151
Fax: (1) 513-576-6104

Australia Operations
Suite 502
655 Pacific Highway
St. Leonards NSW
Australia 2065
Tel: (61) 2 9436 4422
Fax: (61) 2 9436 4355

China Operations
Ceroil Plaza
Room 901/902
9 Yide Street,
Zhongshan Dist.
Dalian 116001, China
Tel: (86) 411 265 6594
Fax: (86) 411 265 6627

European Operations
Bumpers Lane
Sealand Industrial
Estates
Chester
UK CHI 4LT
Tel: (44) 1244 374914
Fax: (44) 1244 379870

India Operations
Plot 1/5 Marol Co-op.
Ind. Est.
Andheri (East)
Mumbai
400 059
Tel: (91-22) 852 2906
Fax: (91-22) 852 1814

©1993 IRD Mechanalysis, Inc.
All rights reserved.

®IRD is a registered trademark of IRD Mechanalysis, Inc.

Subject to change without notice.

FORM #5112

© Copyright 2001 Technical Associates of Charlotte, P.C.

Model 970
Vibration Pickup

SPEC SHEET: 41-06-04
EFFECTIVE: May 1992

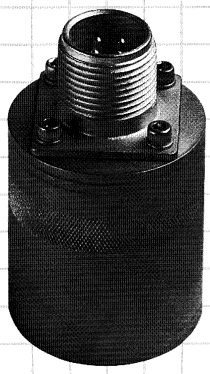
DESCRIPTION

The IRD Model 970 is a rugged low impedance accelerometer type vibration pickup for use with portable instruments on all types of rotating machinery.

The Model 970 has built-in electronics to provide low impedance output and eliminate the need for special low noise cables. The Model 970 is sealed against dust and moisture and operates up to a temperature of 250° F (121°C).

A single 1/4-28 mounting stud is used for attaching the pickup to a flat machine surface. Insulated Mounting Adapters are available for those cases where ground potential problems may exist between the pickup mounting surface and instrument.

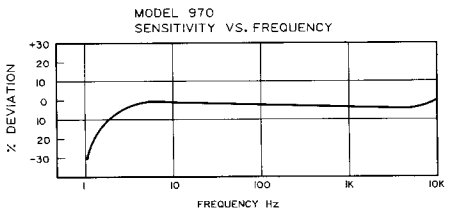
The Model 970 connects directly to IRD Model 810, 840, 880 and other IRD instruments designed to accept the 970. Operating power is obtained from the instrument.



© Copyright 2001 Technical Associates of Charlotte, P.C.

SENSITIVITY
50 mV/g Nominal

FREQUENCY RESPONSE



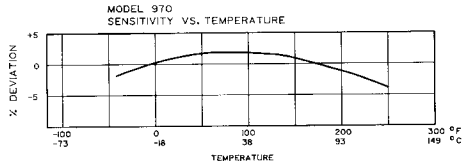
MOUNTED RESONANT FREQUENCY
27K Hz Minimum

MAXIMUM USABLE AMPLITUDE (Undistorted)
180 g peak up to 1 k Hz, 120 g peak at 5K Hz, or 70 g peak at 10 K Hz with ± 15v power supply and 25 ft. of 50 pF/ft. cable. Longer cables may be used with some reduction in maximum amplitude and/or high frequency response.) (Maximum amplitude is reduced proportionally at lower supply voltage)

MAXIMUM VIBRATION (without damage)
1000 g peak sinusoidal vibration

MAXIMUM SHOCK (without damage)
2000 g peak (10 ms half-sine)

TEMPERATURE RESPONSE



ORIENTATION
Any

TRANVERSE RESPONSE
5% Maximum, 3% Typical

MAGNETIC FIELD SENSITIVITY
Less than .001 equiv. g/Gauss at 60 Hz

RESIDUAL NOISE
Less than .0002 equiv. g broadband - transducer output.

GROUNDING
Case isolated from signal ground

POWER SUPPLY
Regulated ± 8 Vdc @ ± 1.2 max max. to ± 15 Vdc @ ± 1.75 ma max.

HOUSING
300 series stainless steel, waterproof, dust proof, welded case

CONNECTOR
4 Pin MS Type 12s-10P

WEIGHT
7.7 oz. (220 gr.)

MOUNTING
1/4-28 Tapped Hole, 1/4" Deep

INSTALLATION NOTES
1. Mounting surface must be flat and smooth, mounting stud must be square with surface and must not exceed 1/4" length.

2. Install with light coating of silicone grease on mating surface, tighten to 30-50 inch-pounds torque.

3. Connector Pin A + DCV power supply, Pin B - DCV power supply, Pin C is signal and power supply common, Pin D signal output.

OPTIONAL ACCESSORIES

- **Insulated Mounting Stud**
- **Magnetic Holders P/N 4332, 24746**
- **Straight Probe P/N 1103.**

ORDERING INFORMATION

(Qty) P/N 19697 Model 970 Vibration Pickup

IRD Mechanalysis - U.S.A.
IRD Mechanalysis, Inc.
6150 Huntley Road
Columbus, OH 43229-9851
Phone: 614/885-5376
Telex: 24-5318
FAX: 614/885-7668

IRD Mechanalysis - Canada
IRD Mechanalysis, Ltd.
P.O. Box 9124
Hamilton, Ontario L8G 3X7
Phone: 416/643-4284
FAX: 416/643-1143

IRD Mechanalysis - U.K.
IRD Mechanalysis (UK), Ltd.
Bumpers Lane
Sealand Industrial Estate
Chester CH1 4LT
Phone: (44) 0244 374914
Telex: 617034
FAX: (44) 0244 379870

IRD Mechanalysis - Belgium
IRD Mechanalysis, N.V.
Excelsiorlaan, 39 Box 2
1930 Zaventem (Zoning Keiberg)
Phone: (2) 721.28.33
Telex: 61256
FAX: (2) 725.00.17

IRD Mechanalysis - India
IRD Mechanalysis, Ltd.
B10, Ansa Industrial Estate
Saki Vihar Road
Saki Naka, Andheri (East)
Bombay 400 072
Phone: (22) 5782906/5780178
Telex: 011-79402
FAX: 0091222027576

IRD Mechanalysis - Australia
IRD Mechanalysis, Pty., Ltd.
337 Pacific Highway
Crows Nest, NSW 2065
Phone: (02) 929 8122
FAX: (02) 922 3937

IRD Mechanalysis - France
IRD Mechanalysis, s.a.r.l.
Z.A. de Courtaboeuf
Miniparc - Batiment 5
6 Avenue des Andes
91952 Les Ulis Cedex
Phone: (16.1) 69.86 11.88
Telex: 603967
FAX: (16.1) 69.86. 10.90

©1992 IRD Mechanalysis, Inc. All rights reserved.

®IRD is a registered trademark of IRD Mechanalysis, Inc.

Subject to change without notice.

L05012

FORM #1853D

© Copyright 2001 Technical Associates of Charlotte, P.C.

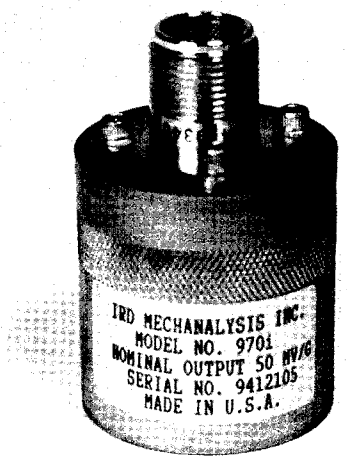
Model 970i

Accelerometer and Accessories

EFFECTIVE: February 1996

DESCRIPTION

The IRD Model 970i is a rugged low impedance accelerometer vibration pickup for use with portable instruments on all types of industrial equipment. The Model 970i has built in electronics to provide a low impedance output and eliminate the need for special low noise cables. The 970i design is based on the proven performance of the IRD Model 970 accelerometer. The 970i provides full database compatibility for users upgrading to newer instrumentation that supports industry standard accelerometers which operate on an instrument supplied constant current DC power source. The Model 970i was designed to provide a very similar frequency response as the standard Model 970 including Spike Energy response. The Model 970i is sealed against dust and moisture and operates up to a temperature of 250° F (121° C).



A single 1/4-28 mounting stud is used for attaching the accelerometer to a flat machine surface. Insulated mounting adapters are available for those cases where ground potential problems may exist between the transducer mounting surface and the instrument.
The Model 970i connects directly to the dataPAC series Data Collectors and any other instruments which operate with industry standard constant current DC power source. Operating power and current for the accelerometer are supplied externally, typically from the instrument.

APPLICATIONS

Well suited as a rugged general purpose portable instrument transducer to measure vibration on a wide variety of machines. The 970i can be used on most machines with one times running speeds between 300 and 15000 RPM (5-250 Hz). The 970i can be temporarily or permanently stud mounted to improve high frequency response.

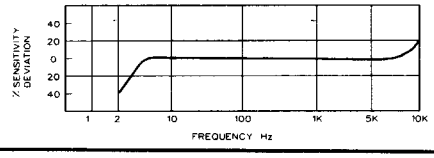
IRD Mechanalysis

© Copyright 2001 Technical Associates of Charlotte, P.C.

TRANSDUCER CHARACTERISTICS

Sensitivity	50 mv/g, ± 10%

Typical Frequency Response



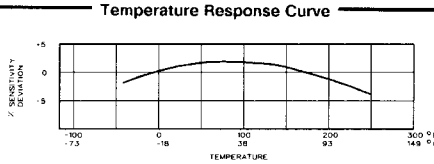
Frequency Response:
± 5% from 6 HZ to 2KHz
± 10% from 5 HZ to 10 KHz
-3dB at 2.5 HZ and + 3 dB at 15 KHz min.

Mounted Resonance	27 KHz
Maximum useable amplitude	75 Gs
(when powered from + 18 to + 24 VDC)	

ENVIRONMENTAL

Maximum Vibration without damage
1000 g peak sinusoidal vibration
Maximum Shock
1000 g peak (10 ms half sine)

Temperature Response Curve



Temperature response	(5% deviation from-30° to + 250° F)
Orientation	Any
Transverse Response	5% maximum, 3% typical
Magnetic Field Sensitivity	Less than .001 equiv. g/Gauss at 60 HZ.

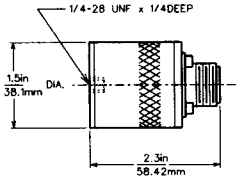
ELECTRICAL

Residual noise	Less than 0.0004 equiv. g broadband - transducer output.
Grounding	Case isolated from signal ground
Power supply	+ 15 to + 24 VDC @ constant current from 2mA to 8mA
Signal output bias voltage	+ 7 to + 11 VDC

PHYSICAL

Housing	300 series stainless steel, waterproof, dustproof, welded case
Connector 2 pin MS type	(Mating connector IRD P/N 4366)
Weight	7.7 oz (220 gr.)
Mounting	1/4-28 Tapped hole, 1/4"depth

Outline Dimension Drawing



Installation Notes

1. Mounting surface must be flat and smooth, mounting stud must be square with surface and must not exceed 1/4" length.
2. Install with light coating of silicone grease on mating surface, tighten to 30 - 50 inch-pounds torque.
3. Connector pin A is signal (+) and connector pin B is return (-).

OPTIONAL ACCESSORIES

Insulated mounting stud - P/N 10308
Magnetic Holders:
Large P/N 4332
Small P/N 24746
Straight Probe P/N 1103

CABLING TO IRD INSTRUMENTS

Any dataPAC series instrument:

- 6ft Coiled Cable - P/N 36960
- 25ft Extension cable (Lemo to Lemo connector) - P/N 37135
- 12ft Straight Cable-BNC connector for Fast Track, 890, 885 and Smart Meter/FS - P/N39480
- 50ft Straight Cable BNC connector for Fast Track, 890, 885 and Smart Meter/FS - P/N 39479
- 6 ft Coiled Cable BNC connector for Fast Track, 890, 885 and Smart Meter/FS - P/N 39450
- 25ft BNC to BNC Extension Cable - P/N 37388

ORDERING INFORMATION

(Qty) P/N 38946 Model 970i Vibration Pickup

ENTEK IRD

www.entekird.com

Corporate Headquarters
1700 Edison Drive
Milford, OH
USA 45150
Tel: (1) 513-576-6151
Fax: (1) 513-576-6104

Australia Operations
Suite 502
655 Pacific Highway
St. Leonards NSW
Australia 2065
Tel: (61) 2 9436 4422
Fax: (61) 2 9436 4355

China Operations
Ceroll Plaza
Room 901/902
9 Yide Street, Zhongshan Dist.
Dalian 116001, China
Tel: (86) 411 265 6594
Fax: (86) 411 265 6627

European Operations
Bumpers Lane
Sealand Industrial Estates
Chester
UK CHI 4LT
Tel: (44) 1244 374914
Fax: (44) 1244 379870

India Operations
Plot 1/5 Marol Co-op. Ind. Est.
Andheri (East)
Mumbai
400 059
Tel: (91-22) 852 2906
Fax: (91-22) 852 1814

© Copyright 2001 Technical Associates of Charlotte, P.C.

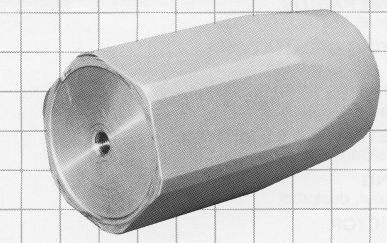
Model 544

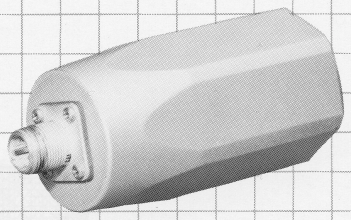
Vibration Pickup - P/N 4526

SPEC SHEET: 41-01
EFFECTIVE: Feb 1990

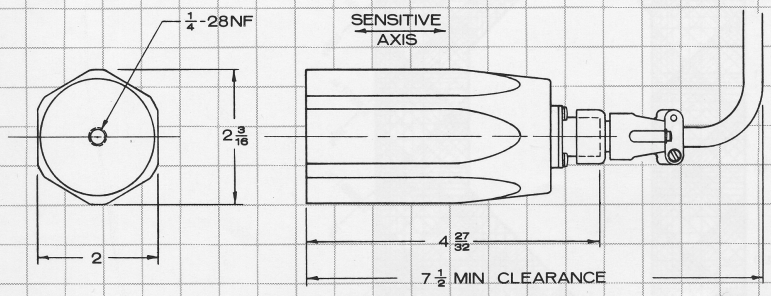
DESCRIPTION

The IRD Model 544 vibration pickup is a seismic velocity transducer (a moving coil in a permanent magnetic field with eddy-current damping). The Model 544 is suitable for a wide variety of manual and continuous monitoring applications due to its high output, low impedance, wide frequency response and rugged construction. The anodized aluminum housing is waterproof, dustproof and acceptable for use in Division 2 hazardous areas. A variety of accessories are available.





OUTLINE DIMENSIONS



INSTALLATION

1. The sensitive axis may be oriented in any direction.
2. The mounting surface must be flat and smooth (minimum diameter = 2-3/8 in).
3. The mounting stud axis must be perpendicular to the mounting surface.
4. Hand-tighten only.
5. Connector pin A is both signal and housing ground; connector pin B is signal ground.

ORDERING INFORMATION

(Qty.) P/N 4526 Model 544 vibration pickup

ENTEK IRD INTERNATIONAL

© Copyright 2001 Technical Associates of Charlotte, P.C.

FREQUENCY RESPONSE
14 to 1,000 Hz within ± 8%; 10 Hz 20% (average) measured at 70° F (21° C)

SENSITIVITY
1080 ± 10% mV/in/s (42.52 mV/mm/s)

TEMPERATURE RANGE
-40 to 500° F (-40 to 260° C)

IMPEDANCE
R = 2 kΩ; L = 0.7 H at 70° F (21° C)

DAMPING FACTOR
0.7 (critical) at 70° F (21° C)

MOUNTING ORIENTATION
Any

TRANSVERSE RESPONSE
Less than 5% (average) 10 to 1,000 Hz

MAGNETIC FIELD SENSITIVITY
0.15 in/s/Gauss

HOUSING
Waterproof, dustproof

CONNECTOR
2-pin (GS-02-10SL-4P)

WEIGHT
21 oz

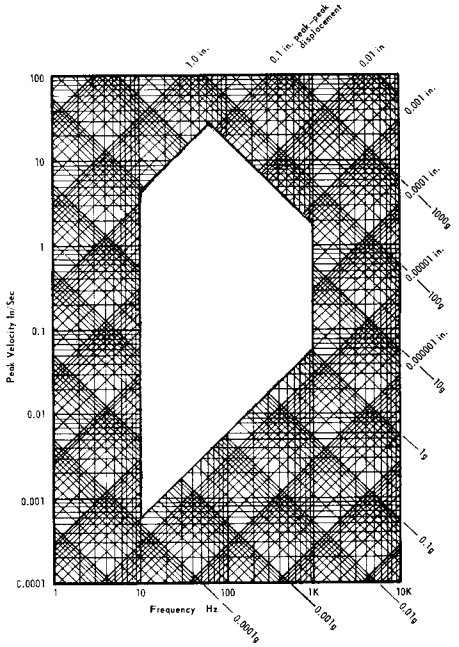
GROUNDING
Case and signal ground common

MOUNTING
1/4 - 28 tapped hole, 1/4 in depth

MAXIMUM AMPLITUDE
125 mils pk-pk to 70 Hz; 30 g above 70 Hz

ACCESSORIES

- Magnetic shields (reduce sensitivity to AC magnetic fields by 100:1) - P/N 10449
- Explosion-proof enclosure (for installation in Division 1, Class I,Group D or Class II, Groups E,F,G hazardous areas) - P/N 4224
- Safety barrier (for intrinsically safe installations in Division 1, Class I, Groups B,C,D hazardous areas) - P/N 7685
- Straight cable connector (rated for continuous operation at 250° F or 121° C) - P/N 4366
- Straight cable connector (rated for continuous operation at 500° F or 260° C) - P/N 15039
- Right-angle connector (rated for continuous operation at 250° F or 121°) - P/N 14797



ENTEK IRD

www.entekird.com

Corporate Headquarters
1700 Edison Drive
Milford, OH
USA 45150
Tel: (1) 513-576-6151
Fax: (1) 513-576-6104

Australia Operations
Suite 502
655 Pacific Highway
St. Leonards NSW
Australia 2065
Tel: (61) 2 9436 4422
Fax: (61) 2 9436 4355

China Operations
Ceroil Plaza
Room 901/902
9 Yide Street,
Zhongshan Dist.
Dalian 116001, China
Tel: (86) 411 265 6594
Fax: (86) 411 265 6627

European Operations
Bumpers Lane
Sealand Industrial Estates
Chester
UK CHI 4LT
Tel: (44) 1244 374914
Fax: (44) 1244 379870

India Operations
Plot 1/5 Marol Co-op. Ind. Est.
Andheri (East)
Mumbai
400 059
Tel: (91-22) 852 2906
Fax: (91-22) 852 1814

05002

FORM # 1249F

© Copyright 2001 Technical Associates of Charlotte, P.C.

Model 560

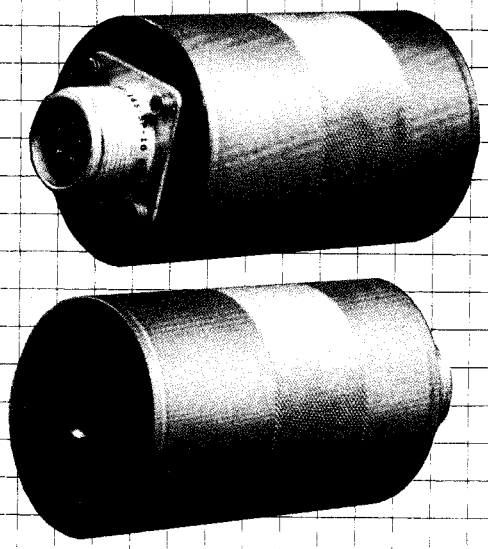
Vibration Pickup P/N 24957

SPEC SHEET: 41-20
EFFECTIVE: Sept 1990

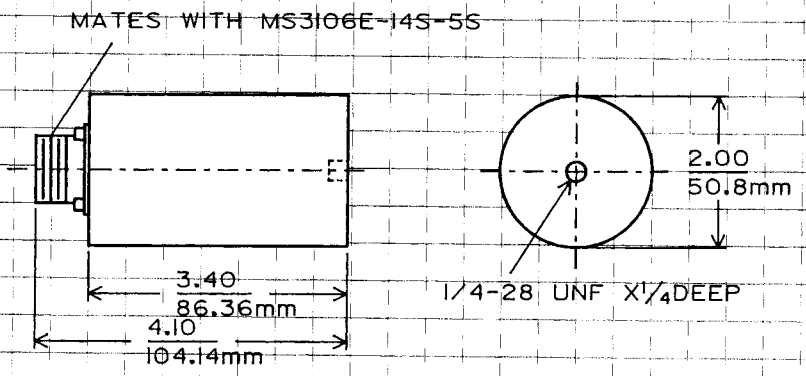
DESCRIPTION

The IRD 560 is a unique piezoelectric seismic velocity transducer with a flat-frequency response from 1 Hz to 4.5 kHz, designed specifically for low-frequency applications. The high upper-frequency limit accommodates both general- and low-frequency machinery vibration measurements. Built-in electronics provide high output sensitivity and low impedance. Low residual noise and low phase shift make it an ideal transducer for low-speed balancing applications.

The IRD 560 is stainless steel and waterproof, with a 1/4-28 tapped hole for stud mounting.



OUTLINE DIMENSIONS



INSTALLATION NOTES:

1. Mounting surface must be flat and smooth.
2. Install on mating surface with light coating of silicone grease.
3. Use 1/4-28 stud mounting for permanent installation; magnetic holder (P/N 4332) for temporary installation.
4. Connector Pin A + DCV power supply; Pin B - DCV power supply; Pin C signal and power supply common; Pin D signal output; Pin E not connected.

ENTEK IRD INTERNATIONAL

© Copyright 2001 Technical Associates of Charlotte, P.C.

SPECIFICATIONS AT +75°F (+24°C)

FREQUENCY RANGE
Flat ±10% (1 Hz to 4.5 kHz)

MAXIMUM USABLE AMPLITUDE (undistorted)
5 in/s (127 mm/s) peak

RESIDUAL NOISE
.0007 in/s (.0178 mm/s) equivalent broad-band

WEIGHT
38 oz (1,079 grams)

SIZE
2 in diameter (50.8 mm)
4 in height (101.6 mm) (including receptacle)

TEMPERATURE RANGE
-40°F to 250°F (-40°C to 121°C)

OUTPUT UNITS
Velocity

VELOCITY SENSITIVITY
1,080 mV/in/s ± 5% at 100 Hz, (764 mVrms/in/s-pk)
42.5 mV/mm/s ± 5% at 100 Hz (30 mVrms/mm/s-pk)

MAGNETIC FIELD SENSITIVITY
10^{-5} in/s/Gauss (2,54 x 10^{-4} mm/s/Gauss equivalent)

TRANSVERSE RESPONSE
5% maximum at 10 Hz, 5 g peak

MAXIMUM CONTINUOUS VIBRATION
250 g pk

MAXIMUM SHOCK
500 g pk

MOUNTING
Centered hole (1/4-28 x 1/4 in deep)

MOUNTED NATURAL FREQUENCY
10 kHz minimum

WATER RESISTANCE
Waterproof

CONNECTOR
5-pin top mounting

GROUNDING
Case isolated from signal ground

POWER INPUT
± 15 Vdc at ± 5 mA maximum

CABLE DRIVE CAPABILITY
No clipping of 1 kHz 5 in/s (127 mm/s) signal over 500 ft (25,000 pF) cable

PHASE SHIFT
10° at 7 Hz, 30° at 3 Hz maximum

ELECTRONICS
Built-in amplifier

ORDERING INFORMATION

(Qty) P/N 24957, IRD 560 Vibration Pickup
(Qty) P/N 24953, Mating Connector
(MS 3106E-14S-5S) for IRD 560
(Qty) P/N 25136, 12 ft cable

ENTEK IRD

www.entekird.com

Corporate Headquarters
1700 Edison Drive
Milford, OH
USA 45150
Tel: (1) 513-576-6151
Fax: (1) 513-576-6104

Australia Operations
Suite 502
655 Pacific Highway
St. Leonards NSW
Australia 2065
Tel: (61) 2 9436 4422
Fax: (61) 2 9436 4355

China Operations
Ceroil Plaza
Room 901/902
9 Yide Street,
Zhongshan Dist.
Dalian 116001, China
Tel: (86) 411 265 6594
Fax: (86) 411 265 6627

European Operations
Bumpers Lane
Sealand Industrial
Estates
Chester
UK CHI 4LT
Tel: (44) 1244 374914
Fax: (44) 1244 379870

India Operations
Plot 175 Marol Co-op.
Ind. Est.
Andheri (East)
Mumbai
400 059
Tel: (91-22) 852 2906
Fax: (91-22) 852 1814

FORM #2008C

© Copyright 2001 Technical Associates of Charlotte, P.C.

Model 580

Vibration Pickup P/N 28852

SPEC SHEET: 41-30
EFFECTIVE: July 1991

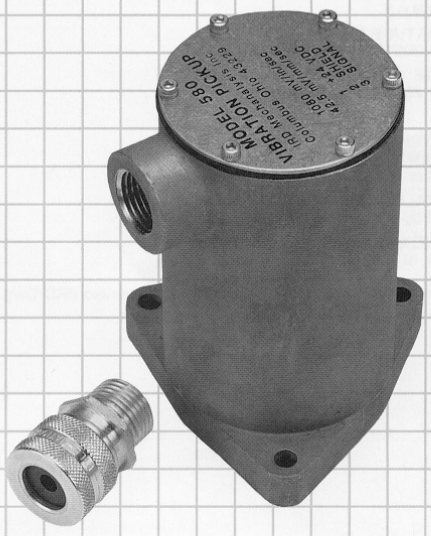
DESCRIPTION

The Model 580 Vibration Pickup is a solid-state, seismic velocity pickup for permanent installation on industrial machinery. It has no moving parts. The waterproof case provides for an exceptionally long, trouble-free service life under adverse conditions, such as in mills, refineries, power plants, and mines. The broad frequency range, 120 to 120,000 CPM makes this pickup suitable for nearly any machine.

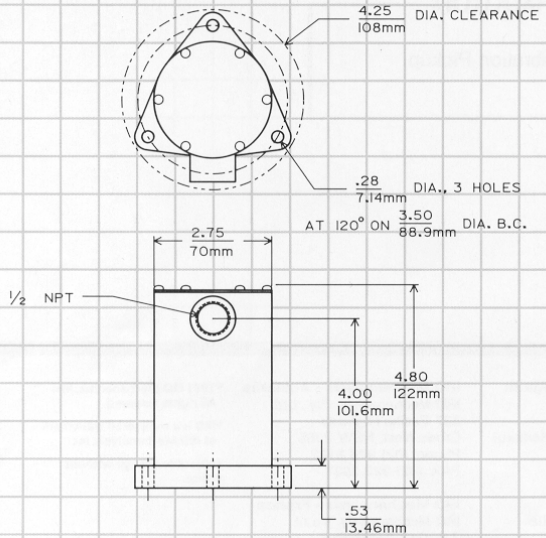
The cast stainless steel housing is designed to withstand mechanical abuse and is resistant to most chemicals. Electrical connections to the pickup are located within the housing protected from the environment. A threaded boss is provided on the case for cable installation through conduit or a sealing gland.

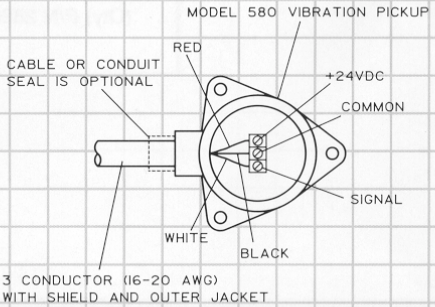
The case is electrically insulated from the internal components, so the Model 580 pickup can be used on insulated bearing machines or where good electrical grounding is not practical.

The Model 580 pickup is Factory-Mutual approved for use in hazardous areas (See Specifications).



OUTLINE DRAWING





NOTE: DO NOT CONNECT SHIELD WIRE. CUT FLUSH WITH CABLE JACKET AND INSULATE.

IRD Mechanalysis

© Copyright 2001 Technical Associates of Charlotte, P.C.

OUTPUT SENSITIVITY
1080mV/in/s (42.5 mV/mm/s)

OUTPUT IMPEDANCE
10 ohm

OUTPUT OFFSET
+8 to +13 Volts D.C.

FREQUENCY RESPONSE
2Hz to 2K Hz (120 to 120,000 CPM), flat within ± 10% @ 70°F.

4Hz to 1.5K Hz (240 to 90,000 CPM), flat within ± 5% @ 70°F.

AMPLITUDE RANGE
0-6 in/s (0-152 mm/s)

ORIENTATION
Omnidirectional

TEMPERATURE RANGE OPERATING
-40° to +200°F. (-40° to + 93°C.)

STORAGE
-55° to +250°F. (-47° to + 121°C.)

SHOCK
500 g peak maximum

WITHSTAND VOLTAGE
400 VDC or peak AC maximum, case to circuit

APPROVALS
Approved by Factory Mutual

INTRINSICALLY SAFE for
Class I, II, III
Division 1
Groups A, B, C, D, E, G when used with safety barriers (IRD Dwg. C98239)

NON-INCENDIVE (Without Safety Barrier) for
Class I, II, III
Division 2
Groups A, B, C, D, & G when installed per (IRD Dwg C98239)

RADIATION
10^7 RAD total dose

MAGNETIC FIELD SENSITIVITY
1.5×10^{-4} in/sec equivalent @ 60 Hz

SEAL
Waterproof to 30 psi (approx. 70 ft. head of water) when installed with waterproof pipe or conduit fittings.

CONSTRUCTION
Type 316 stainless steel case; Neoprene cover gasket, stainless steel hardware.

POWER
+ 18 to + 26 VDC @ 10-mA

CONNECTIONS
Internal 3-screw barrier strip; Clamp terminals accept AWG 16 to 20 wire.

CABLE ENTRY
1/2 NPT threaded boss in case. One (1) compression-type rubber seal fitting included for use with cable up to .250 in. (64 mm) diameter. Seal fitting not for use in radiation or corrosive environments.

DIMENSIONS
4 7/8 in (124 mm) high x 2 3/4 in. (70 mm) body diameter; weight 5.8 lb. (2.46 Kg)

MOUNTING
(3) holes for 1/4 in. (6mm) diameter bolts, 120° apart on 3 1/2 in. (89 mm) diameter circle. Foot print is 4 1/4 in. (108 mm) diameter circle.

ORDERING INFORMATION

(Qty) P/N 28852, Model 580 Vibration Pickup.

IRD Mechanalysis - U.S.A.
IRD Mechanalysis, Inc.
6150 Huntley Road
Columbus, OH 43229-9851
Phone: 614/885-5376
Telex: 24-5318
FAX: 614/885-7668

IRD Mechanalysis - Canada
IRD Mechanalysis, Ltd.
877 Barton Street East
Stoney Creek, Ontario L8G 4S1
Phone: 416/643-4284
FAX: 416/643-1143

IRD Mechanalysis - U.K.
IRD Mechanalysis (UK), Ltd.
Bumpers Lane
Sealand Industrial Estate
Chester CH1 4LT
Phone: (44) 0244 374914
Telex: 617034
FAX: (44) 0244 379870

IRD Mechanalysis - Belgium
IRD Mechanalysis, N.V.
Excelsiorlaan, 39 Box 2
1930 Zaventem (Zoning Keiberg)
Phone: (2) 721.28.33
Telex: 61256
FAX: (2) 725.00.17

IRD Mechanalysis - India
IRD Mechanalysis, Ltd.
B10, Ansa Industrial Estate
Saki Vihar Road
Saki Naka, Andheri (East)
Bombay 400 072
Phone: (22) 5782906/5780178
Telex: 011-79402
FAX: 0091222027576

IRD Mechanalysis - Australia
IRD Mechanalysis, Pty., Ltd.
337 Pacific Highway
Crows Nest, NSW 2065
Phone: (02) 929 8122
FAX: (02) 922 3937

IRD Mechanalysis - France
IRD Mechanalysis, s.a.r.l.
Z.A. de Courtaboeuf
Miniparc - Batiment 5
6 Avenue des Andes
91952 Les Ulis Cedex
Phone: (16.1) 69.86.11.88
Telex: 603967
FAX: (16.1) 69.86.10.90

©1991 IRD Mechanalysis, Inc.
All rights reserved.

*IRD is a registered trademark of IRD Mechanalysis, Inc.

Subject to change without notice.

L05022

FORM #2139C

© Copyright 2001 Technical Associates of Charlotte, P.C.

1900 SERIES PROBES

FEATURES

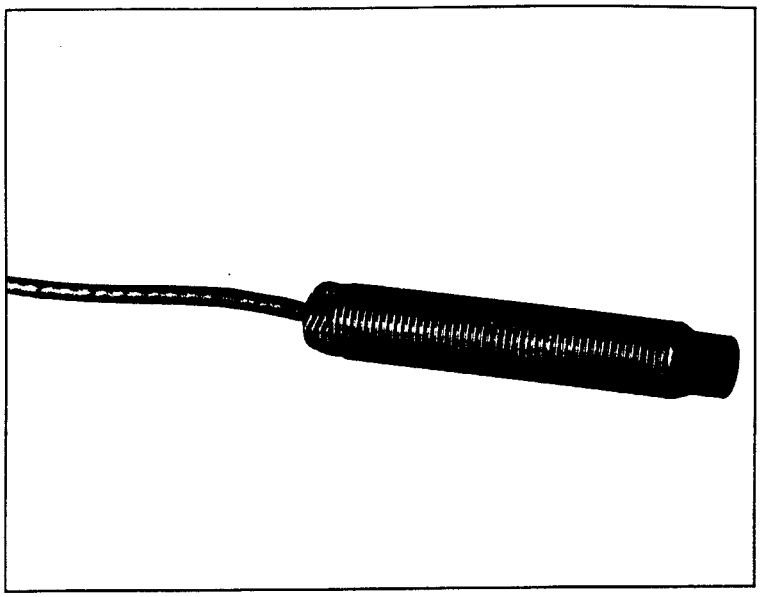
TOTALLY SEALED PROBES

INTEGRAL STAINLESS STEEL WIRE ARMORED CABLE

200°C OPERATION

CABLES LOCKED TO PROBE

HIGH THERMAL SHOCK CAPABILITY



GENERAL OVERVIEW

A probe system consists of an eddy current probe, an oscillator/ demodulator (known as the Probe driver) and an interconnecting cable. These three items form a resonant tuned circuit.

The radio frequency generated by this arrangement is present at the probe tip where a coil radiates a corresponding, low energy, electromagnetic field. This field induces stray electrical (eddy) currents in the surface of the target material placed within its path. The energy loss into the target is detected as a change in the modulated signal and produces a voltage proportional to the gap between the probe tip and the target.

In practical systems the target is the shaft of the machine and the probe is mounted on, or through, the casing. In this way the shaft vibration, relative to the casing, is measured.

The 1900 series of probe systems represents the most significant development of eddy-current probe systems in recent years. They have introduced new standards in terms of probe sealing, cable pull strengths and integrally armored cables, not seen in this type of product in the past.

APPLICATION

The 1900 series are used for measuring single and two plane relative vibration on shafts of rotating machinery. They also find application in measuring shaft eccentricity, axial position of shafts, case expansion movement, differential expansion and other instances where a non-contacting, relative measurement must be made.

MACHINE MOUNTING

Mounting of probes is critical and must be done correctly. The easiest method is to use IRD 29000 probe holders which allow for micrometer adjustment of probes and safe removal of probes whilst machines are running. See the 29000 data sheet for this item.

With each shipment of probe/driver sets, a User Guidance Note (TD-128) is provided which details various methods of installation and suggests methods of avoiding some of the most common difficulties.

PROBE SEALING

To prevent product, lubricant or gases from passing into unwanted areas on the machine being monitored, it is essential that the probe is completely sealed. The IRD 1900 series have set new standards in this area and standard production probes will withstand 100psi (7 bar). The type approval tests included prolonged high pressure tests interspersed with thermal shocks of 220°C.

The probe is sealed into the body and the probe tip sealed to the cable in such a manner that increased pressure tends to produce an improved seal.

© Copyright 2001 Technical Associates of Charlotte, P.C.

CABLE LOCKING

Installation of probes particularly in gas and steam turbines often involves feeding cables through oil ducts or other passage ways. Accidental damage can occur if cables are pulled too hard and industry has seen a history of probes damaged in this way.

The IRD 1900 series have a unique cable which has its steel wire armoring locked into the probe body. This design prevents load being borne by the signal cores, and allows the cable to withstand a continuous pull of 50lbs without damage or change to the probe characteristics.

ARMORED CABLE

The cable used for the IRD 1900 series probes was designed by IRD engineering group especially for this new generation of products. The cable which is only available on IRD probes is a triaxial type with a multi-strand steel center core, an inner screening braid and stainless steel woven armor. The outer sheath is a bright orange PTFE (Teflon) and is quite exceptionally tough. It will not cause failure of the probe if the outer sheath is perforated and the armor is grounded.

Note the exceptionally small bend radius of 12.5 mm which is quite remarkable for a cable of this complexity of construction.

API 670 COMPLIANCE

The IRD probe/driver systems comply fully with the requirements of API 670 and indeed exceed that specification in many areas.

INTRINSIC SAFETY

Full BASEEFA and CENELEC certification is available for the complete range of probes giving certifications to EEx ia IIC T6. See also the section on INTRINSIC SAFETY CERTIFICATIONS; IMPLICATIONS AND APPLICATIONS towards the end of this catalog. Intrinsic safety is of such importance that uncertainty must not exist. IRD are keen to ensure users of its systems fully understand the implications and are supported throughout the design and specification phase of new installations.

PROBE DRIVERS

These are typical of the quality engineering approach of all IRD products. The probe drivers are in stainless steel

TABLE OF PROBE MODELS ETC

This table shows the various models in the range. Essentially they are defined by tip diameter, body thread and type, and whether the cable is fitted with flexible stainless steel conduit. Note the preferred body lengths in the right hand column, and use these if possible. These standard preferred lengths are usually available on a shorter delivery than non standard items. IRD manufacture probes to non standard dimensions for a number of machinery manufacturers, and will be pleased to continue that service. Discuss any special requirements with Technical Sales.

Only specify conduit on probes if really necessary; remembering that the probe cable is stainless steel wire armored, and only needs additional protection outside of the machine, which is generally provided by fixed conduits.

MODEL	TIP DIA	MOUNTING TYPE	THREAD	S/S CONDUIT	PREFERRED BODY LENGTH UNTHREADED	THREADED
1901	5	STD	1/4-28	NO	0	25mm
1902	5	STD	1/4-28	YES	0	25mm
1903	5	STD	M8 x 1	NO	0	25mm
1904	5	STD	M8 x 1	YES	0	25mm
1905	8	STD	3/8-24	NO	0	30mm
1906	8	STD	3/8-24	YES	0	30mm
1907	8	STD	M10 x 1	NO	0	30mm
1908	8	STD	M10 x 1	YES	0	30mm
1909	8	REVERSE	3/8-24	NO	5	30mm
1910	8	REVERSE	3/8-24	YES	5	30mm

© Copyright 2001 Technical Associates of Charlotte, P.C.

castings and are fully encapsulated. They can withstand to 24 hours totally submerged to a depth of 1 meter and will resume working when dried off. IRD however do not recommend this as an operational activity.

SPECIFICATION

5 AND 8 MM PROBES

Signal Sensitivity	200mV/mil
Useful Range	10 - 100 mils
Accuracy	± 5% of gap
Linearity	within 1 mil of best straight line
Voltage at 0.050 gap	10volts ± 1volt
Frequency Response	dc to 10KHz
Temperature Range	-55°C to +200°C
Cable Bend Radius	12.5mm minimum
Cable Pull Strength	50lbs maximum, Straight pull on cable with probe body retained and in line with cable.
Probe Seal	7 bar applied anywhere on the probe.
Connector	Miniature coaxial

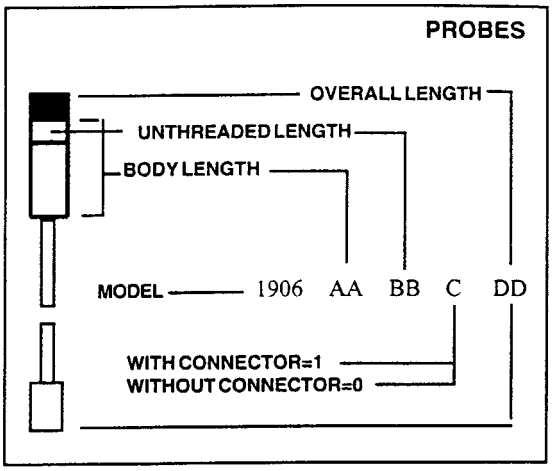
PROBE DRIVERS

Dimensions	To API 670 see illustration
Temperature Range	-55 to +100°C
Construction	Stainless steel casting with encapsulated electronics.
Weight	310g
Waterproofing	Fully encapsulated to withstand 1 meter immersion for 24 hours.

Instrinsic Safety Details
Approved to CENELEC EEx ia IIC T3 - T6
Certificates;
EX 872266X EEx 1a IIC T3 (T.amb = 165°C)
EX 872265X EEx 1a IIC T4 (T.amb = 100°C)
EEx ia IIC T5 (T.amb = 65°C)
EEx ia IIC T6 (T.amb = 50°C)

ORDERING INFORMATION

(See the table on earlier page)



Note that body thread is implicit in model number.

- AA= Body length in mm.(multiples of 5mm**)
- BB= Unthreaded length in mm.(multiples of 5mm**
- C= 1 For fitted connector.
- C= 0 For no connector fitted.
- DD= Total length of probe & cable.

** For notes on preferred standards see table on the previous page.

Select DD to suit the required position of the connector in relation to the conduit arrangements on the machine, but note the preferred standards of 0.5, 1.0 and 1.5 meters.

Probes are fitted with an insulating boot which mates with a matching boot on the Extension cable. These should be pushed together to ensure electrical isolation of the connector body from the ground.

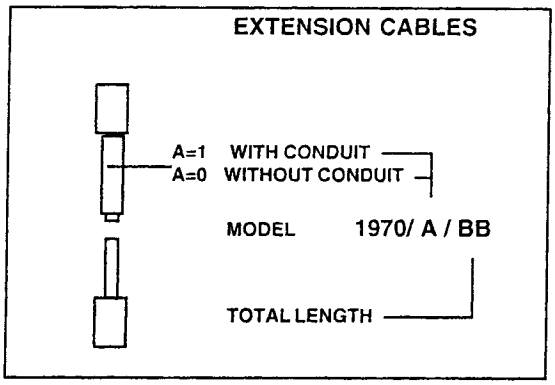
© Copyright 2001 Technical Associates of Charlotte, P.C.

EXTENSION CABLES

When ordering extension cables ensure that the total length of probe and extension cable total 5 or 9 meters, and that this also matches the Probe Driver selection. Preferred lengths of extension cables are therefore 3.5, 4.0, 4.5, 7.5, 8.0, 8.5 meters

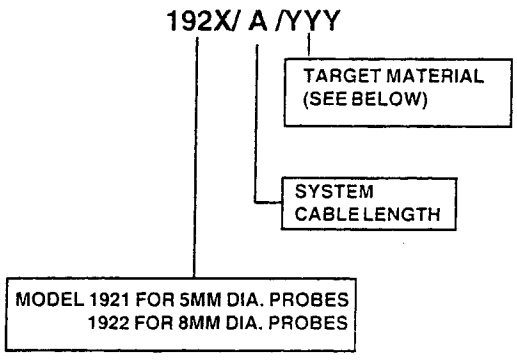
Probes and extension cables are not normally supplied with flexible stainless steel conduit as the cable has an internal stainless steel wire armoring. If flexible stainless steel conduit is preferred to standard rigid conduit on the machine it can be ordered from this section.

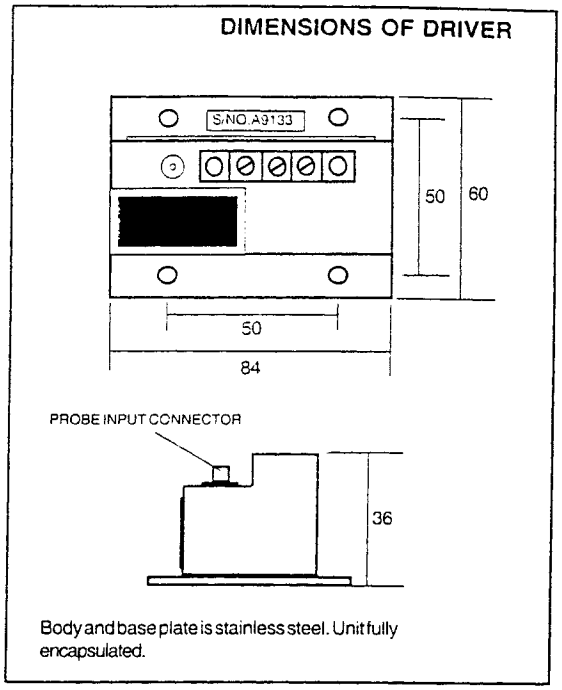
HOWEVER THE PREFERRED METHOD OF PROTECTING CABLES IS TO FIT A PERMANENT CONDUIT SYSTEM ON THE MACHINE AND THE NECESSARY COMPONENTS FOR THIS CAN BE SELECTED FROM THE 'MACHINE MOUNTED STAINLESS STEEL CONDUIT' SECTION IN THIS CATALOG.



PROBE DRIVERS

These are ordered under the following format:





All standard probes are calibrated against a standard target material of En19 (#4140 Steel). Where operation against a specified target material is required, the Probe Driver will be calibrated accordingly.

YYY= 001	EN19 T(#4140 Steel)
002	A-479-410-CL12
003	INCONEL 625
004	EN25
005	EN29A(T) (Use EN40B)
006	EN40B
008	42CMR04 (Use EN19T)
009	BS4360-50 (Use EN14A)
010	BS970 709M40T (Use EN19T)
011	AISA 1045 (Use EN8)
012	BS970 070 M20 (Use EN3B)
013	EN8
014	EN3B
015	EN56B or BS970 420 S29R
016	EN14A
018	EN9

If a material not listed above is to be used, consult Technical Sales for assistance. It may have been added to the list since the publication of this catalog.

A small charge is made for calibrating probe drivers to any material other than En19(#4140 Steel).

© Copyright 2001 Technical Associates of Charlotte, P.C.

© Copyright 2001 Technical Associates of Charlotte, P.C.

Vibra·Metrics® Accelerometer Specifications

APPLICATION		MINIATURE/HIGH FREQUENCY			Triax	GENERAL PURPOSE			LOW COST		Triax	INDUSTRIAL				HIGH TEMPERATURE				UNDER WATER	LOW FREQUENCY		
Specifications (All value typical @ 75 F, 75° RH unless otherwise specified)																							
MODEL		9001A	1000	1018/19	3000	1020S	1022	1026	1136B	1136TB	5302	6036	7000*	7000HG*	8000ST*	1018-6	6036C	8000C*	8000CJ*	1024-1	5102	CBM2000	1030
Sensitivity (A = ± 5%, B = ± 1dB)	mv/g	A: 8.5	A:10	A:10	A:10	A:100	A:100	A:100	B:100	B:100	A:100	A:100	B:100	B:10	A:100	A:8 Pcmb	A:100	A:100	A:10 (at 285°F)	A:100	A:500	A:1500	7000 nom.
Frequency Response	±5% Hz ±3dB Hz	3-20,000 1-30,000	3-15,000 1-40,000	3-10,000 1-30,000	3-10,000 1-11,000	3-10,000 1-12,000	3-5,000 1-10,000	1-5,000 0.5-7,000	3-5k±1dB 1-10,000	3-5k±1dB 1-10,000	3-5,000 1-10,000	3-10,000 1-12,000	3-5k±1db 2-7,000	3-5k±1db 2-7,000	3-5000 2-11,000	3-10,000 1-15,000	3-5,000 1-10,000	3-5,000 2-11,000	30-700 10-2,000	3-10,000 1-11,000	0.5-250 0.2-1000	0.1-500 0.03-1200	0.1-200 0.03-300
Turn-On-Time (2 % of final bias)	Seconds	5	5	5	5	5	5	5	2	2	2	2	0.3	0.3	0.3	N/A	0.3	0.2	0.2	1.5	1.5	1.5	1.5
Amplitude Range (± 1%)	±g(pk)	550	500	500	500	50	50	50	80	80	80	80	80	500	65	–	70	55	550	60	8	3	7
Resonant Frequency	KHz	80	90	60	50	25	25	20	18	18	20	25	18	18	22	60	25	22	22	25	2.3	1.7	0.6
Noise																							
Broadband 2.5 Hz to 25 KHz	µ g (rms)	1500	1500	1500	1500	250	200	200	250	250	250	250	300	3000	300	–	250	25	1500	250	40	50	20
Spectral 10 Hz	µ g√Hz	100	100	100	100	14	14	14	12	12	12	12	28	280	26	–	12	8	200	12	4.0	1.0	0.6
100 Hz	µ g√Hz	25	25	25	25	4	4	4	5	5	5	5	8	80	8	–	5	4	40	5	2.0	0.2	0.2
1000 Hz	µ g√Hz	10	10	10	10	3	3	3	2	2	2	2	4	40	4	–	2	2	15	2	1.5	0.08	0.06
Transverse Sensitivity (typical)		<5%	<5%	<5%	<5%	<5%	<5%	<5%	<7%	<7%	<5%	<5%	<7%	<7%	<5%	<5%	<5%	<5%	<5%	<5%	<5%	<5%	<5%
Output Impedance	OHM	1000	1000	1000	1000	1000	1000	1000	100	100	100	1000	1000	1000	1000	–	1000	1000	1000	1000	1000	1000	1000
EMI/RFI Protection	dB	20	52	52/20	52	52	52	52	52	52	52	80	52	52	52	52	80	52	52	80	52	52	52
ESD Protection		yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes
Ground Isolation	M ohm	–	100	100/–	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100
Bias Volts (nom.)		7	7	7	7	7	7	7	12	12	12	12	12	12	12	–	10	8	10	10	7	7	7
Power	Vdc, 2-4ma	15-30	15-30	15-30	15-30	15-30	15-30	15-30	24-30	24-30	24-30	24-30	24-30	24-30	24-30	–	24-30	20-30	24-30	24-30	15-30	15-30	15-30
Temperature Operating	(°C/°F)																						
Minimum		-40/-40	-40/-40	-40/-40	-40/-40	-40/-40	-40/-40	-40/-40	-20/-10	-20/-10	-40/-40	-40/-40	-20/-10	-20/-10	-40/-40	-40/-40	-40/-40	-40/-40	-40/-40	-40/-40	-40/-40	-40/-40	-40/-40
Maximum		121/250	121/250	121/250	121/250	121/250	121/250	121/250	95/200	95/200	121/250	121/250	95/200	95/200	121/250	180/350	180/350	180/350	180/350	121/250	121/250	121/250	121/250
Size height (in/cm)		.30/.8	.45/1.1	.86/2.2	1.1/2.7	1.2/3.0	1.2/3.0	1.2/3.0	1.5/3.8	2.3/5.8	1.5x1.7x1.7	2.0/5.1	.85/2.1	.85/2.1	.85/2.1	1.1/2.8	2.4/6.1	.85/2.1	.85/2.1	1.7/4.3	3/7.6	2.4x2.2x2.2	2.5/6.4
flats (in/cm)		.25/.6	.3/.7	.5/1.3	1.0/2.5	.75/1.9	.75/1.9	.75/1.9	1.0.2.5	1.0/2.5	3.8x4.3x4.3	.9/2.2	1.0/2.5	1.0/2.5	1.0/2.5	.6/1.5	9/2.2	1.0/2.5	1.0/2.5	.8/2.0	1.3/3.3	6.1x5.6x5.6	2/5.1
Weight grams (without cable)		.7	2	16	10	35	50	23	55	90	189	90	30	30	30	28	110	30	30	60	252	1250	430
Seal		epoxy	epoxy	epoxy	epoxy	epoxy	epoxy	epoxy	hermetic	hermetic	hermetic	welded	welded	welded	welded	epoxy	welded	welded	welded	welded	welded	hermetic	epoxy
Connector (S=Side, T=Top)		S, cable to 10-32	S, cable to 10-32	S, 10-32	S, 4-Pin	S, 10-32	S, cable to 1/4-28	S, 10-32	S, cable to BNC	T, cable to BNC	T, 4-Pin	T, 2-Pin	S, cable to BNC	S, cable to BNC	S, cable to BNC	S, 10-32	T, 2-Pin	S, cable to BNC	S, cable to BNC	S, cable 15' to PT	T, 2-Pin	S, 2-Pin	S, Coax
Mounting Stud		None	6-32	10-32	10-32	1/4-28	1/4-28	10-32	1/4-28	1/4-28	1/4-28	1/4-28	10-32	10-32	10-32	10-32	1/4-28	10-32	10-32	1/4-28	1/4-28	2x10-32	1/4-28
Design		iso. comprs	iso. comprs	iso. comprs	iso. comprs	iso. comprs	iso. comprs	iso. comprs	iso. comprs	iso. comprs	iso. comprs	iso. comprs	shear	shear	shear	iso. comprs	iso. comprs	shear	shear	iso. comprs	shear	shear	diaphragm
Sensor Highway® Compatible		yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes
Other Versions Available		• 9002A Low Noise • 100 mv/g	• Case Gnd • 50 KHz • Shock		• 3000X 7 KHz				• FM Intrinsically safe • Low Bias			• FM Intrinsically safe			• FM Intrinsically safe • TUFKEM Cable			• TUFKEM Cable	• TUFKEM Cable		• 5100 w/TNC		

1. As a part of continual product improvement, specifications subject to change without notice
2. Registered trademarks of Vibra-Metrics, Inc. include: Sensor Highway, IsoShield, TUFKEM.
3. Calibration supplied with accelerometer is 10-10 KHz
** Available on selected models. Consult factory for details

Bulletin No. VM17-2

LIFETIME WARRANTY ON 8000 SERIES**

* Patent No. 5,473,941

Vibra·Metrics, Inc.
1014 Sherman Avenue · Hamden, CT 06514 · Tel (203) 288-6158 · Fax (203) 288-4937

FOR APPLICATIONS ASSISTANCE OR ORDER PLACEMENT CALL OUR HOT LINE 1-800 U-SENS-IT

INDUSTRIAL MONITORING INSTRUMENTATION
A division of PCB Piezotronics, Inc.

INDUSTRIAL ICP® ACCELEROMETER SPECIFICATIONS

Model No. **326A01**

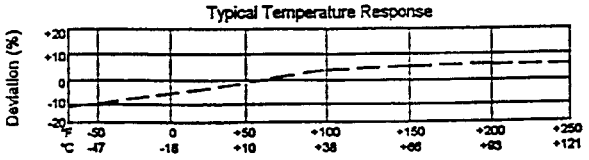
Revisions: -B- Rev # 4488

	ENGLISH	S.I.
DYNAMICS		
Sensitivity (± 5 %) [A]	100 mV/g	10,2 mV/m/s²
Measurement Range	±50 g	±490 m/s²
Resolution	0.0001 g	0,001 m/s²
Frequency Range: (± 5 %) [B]	30 - 300,000 cpm (0.5 - 5,000 Hz)	30 - 300 000 cpm (0,5 - 5 000 Hz)
(±10%)	18 - 420,000 cpm (0.3 - 7,000 Hz)	18 - 420 000 cpm (0,3 - 7 000 Hz)
(±3dB)	12 - 600,000 cpm (0.2 - 10,000 Hz)	12 - 600 000 cpm (0,2 - 10 000 Hz)
Resonant Frequency [C]	1,200 kcpm (20 kHz)	1 200 kcpm (20 kHz)
Transverse Sensitivity	≤5 %	≤5 %
Amplitude Linearity [D]	±1 %	±1 %
ENVIRONMENTAL		
Shock Limit (peak)	5,000 g	49 050 m/s²
Temperature Range	- 65° to +250°F	- 54° to +121°C
Temperature Coefficient	See Graph	See Graph
Strain Sensitivity	0.0005 g/με	0,0049 m/s²/με
ELECTRICAL		
Settling Time [E]	60 sec	60 sec
Excitation Voltage	18 to 28 volts	18 to 28 volts
Excitation Current	2 to 20 mA	2 to 20 mA
Electrical Noise Broadband	100 μg	981 μm/s²
Electrical Noise Spectrical: 1 Hz	10.0 μg/√Hz	98,0 (μm/s²)/√Hz
10 Hz	2.0 μg/√Hz	20,0 (μm/s²)/√Hz
100 Hz	0.6 μg/√Hz	6,0 (μm/s²)/√Hz
1000 Hz	0.2 μg/√Hz	2,0 (μm/s²)/√Hz
Output Impedance	<100 ohms	<100 ohms
Output Bias	8 to 14 volts	8 to 14 volts
Full Scale Output Voltage [F]	±5 volts	±5 volts
Discharge Time Constant	≥1.0 sec	≥1,0 sec
Case Isolation	≥10⁸ ohms	≥10⁸ ohms
Protection	RFI / ESD	RFI / ESD
MECHANICAL		
Size: Hex x Height	1 3/16 x 2 3/16	30,2 x 55,6 mm
Weight	5.3 oz	150 gm
Mounting Thread (Female) [G]	1/4 - 28	Not Applicable
Mounting Torque	2 to 5 ft-lb	2,7 to 6,8 N-m
Sensing Element	Shear - Ceramic	Shear - Ceramic
Case Material	316L St. Stl.	316L St. Stl.
Sealing	Welded Hermetic	Welded Hermetic
Connector Type (2 - Pin)	Mil-C-5015	Mil-C-5015
Connector Position	Top	Top
Mating Connector	MS 3106	MS 3106

Supplied Accessories:
Model 081B20 Mounting Stud
Model 085A31 Protective/Thermal Jacket
Calibration Certificate (10 to 5,000 Hz)

Note:
[A] Conversion Factor: 1g = 9,81 m/s²
[B] 1 Hz = 60 cpm (cycles per minute)
[C] Mounted Resonance (nominal)
[D] Zero Base Best Straight Line
[E] Within 1% of output bias
[F] For FSO at 14V bias, minimum of 20V excitation is required
[G] 1/4-28 thread has no equivalent in SI system

Typical Temperature Response



Specifications subject to change without notification.

App'd: U.F. 9/11/97	Engr: 9/11/97	Sales: 9/11/97	Spec No. 326-1010-80

© Copyright 2001 Technical Associates of Charlotte, P.C.

INDUSTRIAL MONITORING INSTRUMENTATION
A division of PCB PIEZOTRONICS, INC.

INDUSTRIAL ICP® ACCELEROMETER
SPECIFICATIONS

Model No.
326A02

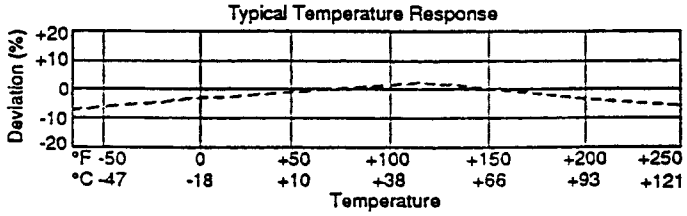
Revision:
-A- Rev #4031

	ENGLISH	S.I.
DYNAMICS		
Sensitivity (±5%) [A]	500 mV/g	51 mV/m/s²
Measurement Range	±10 g	±98 m/s²
Resolution	0.00001 g	0,0001 m/s²
Frequency Range: (±5%) [B]	30 - 120,000 cpm (0.5 - 2,000 Hz)	30 - 120000 cpm (0,5 - 2000 Hz)
(±10%)	18 - 240,000 cpm (0.3 - 4,000 Hz)	18 - 240000 cpm (0,3 - 4000 Hz)
(± 3dB)	12 - 360,000 cpm (0.2 - 6,000 Hz)	12 - 360000 cpm (0,2 - 6000 Hz)
Resonant Frequency [C]	600 kcpm (10 kHz)	600 kcpm (10 kHz)
Transverse Sensitivity	≤7 %	≤7 %
Amplitude Linearity [D]	≤1 %	≤1 %
ENVIRONMENTAL		
Shock Limit (Peak)	5,000 g	49050 m/s²
Temperature Range	- 65 to +250 °F	- 54 to +121 °C
Temperature Coefficient	See Graph	See Graph
Strain Sensitivity	0.0005 g/με	0,0049 m/s²/με
ELECTRICAL		
Settling Time [E]	60 sec	60 sec
Excitation Voltage	18 to 28 volts	18 to 28 volts
Excitation Current	2 to 20 mA	2 to 20 mA
Electrical Noise Broadband	10 μg/√Hz	98 (μm/s²)/√Hz
Electrical Noise Spectral: (1 Hz)	2.0 μg/√Hz	20 (μm/s²)/√Hz
(10 Hz)	0.3 μg/√Hz	2,9 (μm/s²)/√Hz
(100 Hz)	0.2 μg/√Hz	2,0 (μm/s²)/√Hz
(1,000 Hz)	0.1 μg/√Hz	1,0 (μm/s²)/√Hz
Output Impedance	<100 ohms	<100 ohms
Output Bias	8 to 11 volts	8 to 11 volts
Full Scale Output Voltage	±5 volts	±5 volts
Discharge Time Constant	≥1.0 sec	≥1,0 sec
Case Isolation	≥10⁸ ohms	≥10⁸ ohms
Protection	RFI / ESD	RFI / ESD
MECHANICAL		
Size: Hex x Height	1 3/16 x 2 3/16 in	30,2 x 55,6 mm
Weight	7.4 oz	210 gm
Mounting Thread (Female) [F]	1/4-28	Not Applicable
Mounting Torque	2 to 5 ft-lb	2,7 to 6,8 N-m
Sensing Element	Ceramic	Ceramic
Case Material	316L St. Stl.	316L St. Stl.
Sealing	Hermetic	Hermetic
Connector Type (2-pin)	Mil-C-5015	Mil-C-5015
Connector Position	Top	Top
Mating Connector	MS3106	MS3106

Supplied with Accelerometer:
Model 081B20 Mounting Stud
Model 085A31 Protective/Thermal Jacket
Calibration Certificate (10 to 2,000 Hz)

Notes:
[A] Conversion Factor: 1g = 9,81 m/s²
[B] 1 Hz = 60 cpm (cycles per minute)
[C] Mounted Resonance (nominal)
[D] Zero Based, Best Straight Line
[E] Within 1% of Output Bias
[F] 1/4-28 has no equivalent in S.I.units.

Typical Temperature Response



Specifications subject to change without notification.

App'd:	Engr:	Sales:	Spec. No.
11/7/93	11/7/93	11/7/93	326-1020-80

© Copyright 2001 Technical Associates of Charlotte, P.C.

INDUSTRIAL MONITORING INSTRUMENTATION
A division of PCB PIEZOTRONICS, INC.

INDUSTRIAL ICP® ACCELEROMETER
SPECIFICATIONS

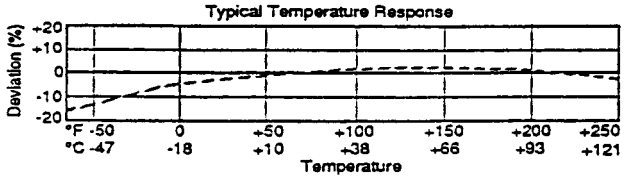
Model No.
326A03
Revision:

		ENGLISH	S.I.
DYNAMICS			
Sensitivity (±5%) [A]		1000 mV/g	102 mV/m/s²
Measurement Range		±5 g	±49 m/s²
Resolution		0.000005 g	0,000049 m/s²
Frequency Range: (±5%) [B]		30 - 120,000 cpm (0.5 - 2,000 Hz)	30 - 120 000 cpm (0,5 - 2 000 Hz)
	(±10%)	18 - 240,000 cpm (0.3 - 4,000 Hz)	18 - 240 000 cpm (0,3 - 4 000 Hz)
	(± 3dB)	12 - 360,000 cpm (0.2 - 6,000 Hz)	12 - 360 000 cpm (0,2 - 6 000 Hz)
Resonant Frequency [C]		600 kcpm (10 kHz)	600 kcpm (10 kHz)
Transverse Sensitivity		≤7 %	≤7 %
Amplitude Linearity [D]		≤1 %	≤1 %
ENVIRONMENTAL			
Shock Limit (Peak)		5,000 g	49 050 m/s²
Temperature Range		- 65 to +250 °F	- 54 to +121 °C
Temperature Coefficient		See Graph	See Graph
Strain Sensitivity		0.0005g/με	0,0049 m/s²/με
ELECTRICAL			
Settling Time [E]		60 sec	60 sec
Excitation Voltage		18 to 28 volts	18 to 28 volts
Excitation Current		2 to 20 mA	2 to 20 mA
Electrical Noise Broadband		5.0 μg/√Hz	49 (μm/s²)/√Hz
Electrical Noise Spectral:	(1 Hz)	1.0 μg/√Hz	10 (μm/s²)/√Hz
	(10 Hz)	0.2 μg/√Hz	2,0 (μm/s²)/√Hz
	(100 Hz)	0.1 μg/√Hz	1,0 (μm/s²)/√Hz
	(1,000 Hz)	0.05 μg/√Hz	0,5 (μm/s²)/√Hz
Output Impedance		<100 ohms	<100 ohms
Output Bias		8 to 11 volts	8 to 11 volts
Full Scale Output Voltage		±5 volts	±5 volts
Discharge Time Constant		≥1.0 sec	≥1,0 sec
Case Isolation		≥10⁸ ohms	≥10⁸ ohms
Protection		RFI / ESD	RFI / ESD
MECHANICAL			
Size: Hex x Height		1 3/16 x 2 3/16 in	30,2 x 55,6 mm
Weight		7.4 oz	210 gm
Mounting Thread (Female) [F]		1/4-28	Not Applicable
Mounting Torque		2 to 5 ft-lb	2,7 to 6,8 N-m
Sensing Element		(Ceramic)	Ceramic
Case Material		316L St. Stl.	316L St. Stl.
Sealing		(Hermetic)	Hermetic
Connector Type (2-pin)		Mil-C-5015	Mil-C-5015
Connector Position		Top	Top
Mating Connector		MS3106	MS3106

Supplied with Accelerometer:
Model 081B20 Mounting Stud
Model 085A31 Protective /Thermal Jacket
Calibration Certificate (10 to 2,000 Hz)

Notes:
[A] Conversion Factor: 1g = 9,81 m/s²
[B] 1 Hz = 60 cpm (cycles per minute)
[C] Mounted Resonance (nominal)
[D] Zero Based, Best Straight Line
[E] Within 1% of Output Bias
[F] 1/4-28 has no equivalent in S.I.units.

Typical Temperature Response



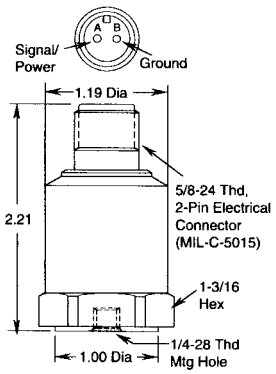
Specifications subject to change without notification.

App'd:	Engr:	Sales:	Spec. No.
D. LALLY 1/6/93	1/6/93	1/6/93	326-1030-80

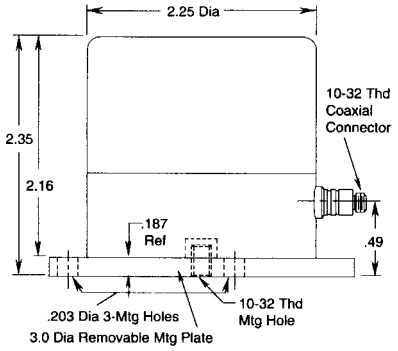
© Copyright 2001 Technical Associates of Charlotte, P.C.

Dimensions shown in inches except where noted.

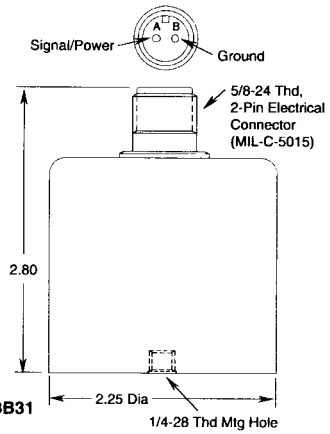
English Selection Guide



393A03 & 393B12



393C



393B31

		GENERAL PURPOSE		HIGH SENSITIVITY	
MODEL NUMBER	**Unit**	**393A03**	**393C**	**393B12**	**393B31**
Voltage Sensitivity (1)	mV/g	1 000	1 000	10 000	10 000
Frequency Range (±5%)	Hz	0.5 to 2 000	0.025 to 800	0.15 to 1000	0.1 to 200
(±10%)	Hz	0.3 to 4 000	0.01 to 1 200	0.1 to 1500	0.07 to 300
Resonant Frequency	kHz	≥ 10	≥ 3.5	≥ 10	≥ 1.0
Amplitude Range	±g pk	5	1.5	0.5	0.5
Resolution (broadband)	g pk	0.000005	0.0001	0.000005	0.000001
Mechanical Shock Limits	±g pk	5 000	100	200	40
Temperature Range	°F	-65 to +250	-65 to +200	-50 to +180	0 to +150
Temperature Coefficient	%/°F	≤ 0.05	≤ 0.03	≤ 0.05	≤ 0.08
Amplitude Linearity	%	± 1	± 1	± 1	± 1
Transverse Sensitivity	%	≤ 5	≤ 5	≤ 5	≤ 5
Base Strain Sensitivity	g/με	≤ 0.0005	≤ 0.001	≤ 0.0005	≤ 0.005
Excitation Voltage	VDC	20 to 30	18 to 30	20 to 30	24 to 28
Constant Current Excitation	mA	2 to 20	2 to 20	2 to 20	2 to 10
Output Impedance	ohm	≤ 200	≤ 300	≤ 300	≤ 500
Output Bias Voltage	VDC	8 to 14	3 to 6	8 to 14	8 to 14
Discharge Time Constant	second	≥ 1	≥ 20	≥ 3.5	≥ 5
Ground Isolation (2)	ohm	$\geq 10^8$	$\geq 10^8$	$\geq 10^8$	$\geq 10^8$
Sensing Element	type	ceramic	quartz	ceramic	ceramic
Element Configuration	structure	tri-shear	compression	tri-shear	flex-tensional
Connector	type	Mil-C-5015	10-32 coax	Mil-C-5015	Mil-C-5015
Sealing	type	weld/herm	weld/herm	weld/herm	weld/herm
Mounting Thread (3)	size	1/4-28	10-32	1/4-28	1/4-28
Housing	material	stainless stl	stainless stl	stainless stl	stainless stl
Weight	gram (oz)	210 (7.4)	1 000 (35.3)	200 (7.05)	635 (22.4)
SUPPLIED ACCESSORIES: (3, 4, 5)					
Mounting Stud (6)		081B20	081B05	081B20	081B20
Protective / Thermal Boot (7)		085A31	N/A	085A31	N/A
OPTIONAL ACCESSORIES: (4)					
Magnetic Mounting Base		080A54	080A21	080A54	N/A
Triaxial Mounting Adaptor		080A57	080M16	080A57	N/A
CABLING: (4)					
Mating Cable Connectors	type	E, M, P	A, H, K, W	E, M, P	E, M, P
Recommended Stock Cable	series	002, 012, 024	002	002, 012, 024	002, 012, 024

NOTES:

1. Models 393A03 and 393B31 have a sensitivity tolerance of ±5%. Models 393C and 393B12 have a sensitivity tolerance of ±10%.
2. Model 393C is base isolated; remaining models utilize case isolated design.
3. Model 393C also supplied with Model 080A88 three hole mounting flange.
4. See page cables/accessories section beginning on page 78 for a complete description of the supplied and optional models.
5. NIST traceable calibration certificate from 10 Hz to +5% frequency point supplied with each accelerometer.
6. Model 081B05 mounting stud is threaded 10-32 to 10-32. Model 081B20 mounting stud is threaded 1/4-28 to 1/4-28.
7. Protective / thermal boot reduces thermal transient errors at low frequencies.

© Copyright 2001 Technical Associates of Charlotte, P.C.

Piezoelectric Acceleration Pickups

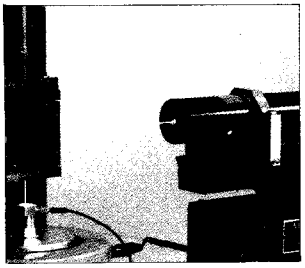
High performance pickups must be used to measure vibration accurately. The piezoelectric acceleration pickup is small, lightweight and can even be used for measuring high frequency vibrations. When the object to be measured is small or if the object is a machine part, the vibration measurement must be made with this type of pickup. RION products various models of acceleration pickups ranging in size from miniature to large output models including triaxial models.

Model	PV-32A	PV-34	PV-35	PV-36	PV-37A	PV-44
APPEARANCE						
WEIGHT, g	33	36	30	19	18	29
DIMENSIONS, mm (Hex x H)	25 x 25 x 25 (W x D x H)	17 x 22	17 x 23	14 x 20	14 x 19	16 x 20.5
STRUCTURE	Compression	Compression	Compression	Compression	Compression	Compression
CHARGE SENSITIVITY, PC/G	13	88	85	17	15	75
VOLTAGE SENSITIVITY, mV/G	11	80	77	15	14	70
CAPACITANCE, pF (cable incl.)	1,100	1,100	1,100	1,100	1,100	1,100
FREQUENCY RESPONCE, Hz (±1.0 dB)	1~5,000	1~8,000	1~8,000	1~15,000	1~8,000	1~10,000
MOUNTED RESONANCE FREQUENCY, kHz	20	30	30	50	30	28
WORKING TEMPERATURE RANGE, °C	-50~+160	-50~+160	-50~+160	-50~+160	-50~+160	-50~+260
ACCELERATION RANGE, G	400	500	500	500	500	400
CABLE & CONNECTOR TYPE	VP-51C	VP-51A	VP-51A	VP-51A	VP-51A	VP-51B
MOUNTING	M6 Stud	M6 Stud	M6 Stud	M6 Stud	M6 Stud	10-32 Stud

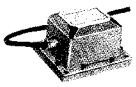
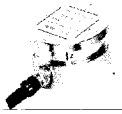
Model	PV-65	PV-81	PV-84	PV-85	PV-86	PV-87
APPEARANCE						
WEIGHT, g	26	405	80	23	23	115
DIMENSIONS, mm (Hex x H)	16 x 22.5	42 x 42 x 42 (W x D x H)	24 x 25	17 x 18.5	17 x 18.5	24 x 30.5
STRUCTURE	Shear	Shear	Shear	Shear	Shear	Shear
CHARGE SENSITIVITY, PC/G	60	1,000	160	58	58	320
VOLTAGE SENSITIVITY, mV/G	17	500	170	64	64	350
CAPACITANCE, pF (cable incl.)	3,700	2,000	900	900	900	900
FREQUENCY RESPONCE, Hz (±1.0 dB)	1~10,000	0.5~6,000	1~4,000	1~8,000	1~8,000	1~3,000
MOUNTED RESONANCE FREQUENCY, kHz	23	4	13	22	20	7
WORKING TEMPERATURE RANGE, °C	-50~+260	-50~+160	-50~+160	-50~+160	-50~+160	-50~+160
ACCELERATION RANGE, G	400	(40)	200	500	500	100
CABLE & CONNECTOR TYPE	VP-51B	VP-51A	VP-51A	VP-51A	VP-51A	VP-51A
MOUNTING	M6 Stud	M6 Stud	M6 Stud	M6 Stud	M6 Stud	M6 Stud

© Copyright 2001 Technical Associates of Charlotte, P.C.

Model	PV-90A	PV-92	PV-93	PV-95
APPEARANCE				
WEIGHT, g	1.2	6.3	28	9
DIMENSIONS, mm (Hex x H)	6 x 10	19.5 x 11 x 7 (W x D x H)	21 x 21x 16 (W x D x H)	14 x 15
STRUCTURE	Shear	Shear	Shear	Shear
CHARGE SENSITIVITY, PC/G	1.8	2.6	5.4	5.5
VOLTAGE SENSITIVITY, mV/G	2.4	3.5	8	8
CAPACITANCE, pF (cable incl.)	750	750	670	670
FREQUENCY RESPONCE, Hz (±1.0 dB)	1~25,000	1~25,000	1~8,000	1~10,000
MOUNTED RESONANCE FREQUENCY, kHz	70	70	45	50
WORKING TEMPERATURE RANGE' °C	-50~+160	-50~+160	-50~+160	-50~+160
ACCELERATION RANGE, G	500	500	1,000	1,000
CABLE & CONNECTOR TYPE	VP-51D	VP-51D x 3	VP-51C	VP-51A
MOUNTING	M3 Stud	Adhesive	M6 Stud	M6 Stud



Vibration pickup Calibration

Model	PV-20	PV-22	PV-62	PV-83A	PV-88A
APPEARANCE					
STRUCTURE	Shear	Shear	Shear	Shear	Shear
VOLTAGE SENSITIVITY, V/1,000 Ga1	1	5	*35 mV/G	0.6	0.6
FREQUENCY RESPONSE, Hz	0.1~1,000	0.1~30	1~100	1~90	1~90
ACCELERATION RANGE, Ga1	1~5,000	0.3~1,000	0.3~3,000	0.03~3,000	0.03~3,000
WORKING TEMPERATURE RANGE, °C	0~+40	0~+40	-20~+60	0~+40	0~+40
WEIGHT, g	560	2,500	400	740	430
DIMENSIONS, mm	60Wx98Dx25H	70 Dia. x 300	210 Dia. x 12	84 Dia. x 61	64 Dia. x 58
POWER SUPPLY	±6 VDC	±6 VDC	–	9 VDC	9 VDC
WITH INTERNAL ELECTRONICS	Charge Amplifier 3-direction	Charge Amplifier 3-direction	NO *Charge Sensitivity (30 pC/G) 3-direction	Pre Amplifier Low Pass Filter 3-direction	Pre Amplifier Low Pass Filter 1-direction

© Copyright 2001 Technical Associates of Charlotte, P.C.

Vibra·Metrics® Accelerometer Specifications

APPLICATION	MINIATURE/HIGH FREQUENCY			Triax	GENERAL PURPOSE			LOW COST		Triax	INDUSTRIAL				HIGH TEMPERATURE				UNDER WATER	LOW FREQUENCY		
Specifications (All value typical @ 75 F, 75% RH, unless otherwise specified.) MODEL	9001A	1000	1018/19	3000	1020S	1022	1026	1136B	1136TB	5302	6036	7000*	7000HG*	8000ST*	1018-6	6036C	8000C*	8000CJ*	1024-1	5102	CBM2000	1030
Sensitivity (A = ± 5%, B = ± 1dB) mv/g	A: 8.5	A:10	A:10	A:10	A:100	A:100	A:100	B:100	B:100	A:100	A:100	B:100	B:10	A:100	A:8 Pcmb	A:100	A:100	A:10 (at 285°F)	A:100	A:500	A:1500	7000 nom
Frequency Response ±5% Hz	3-20,000	3-15,000	3-10,000	3-10,000	3-10,000	3-5,000	1-5,000	3-5k±1dB	3-5k±1dB	3-5,000	3-10,000	3-5k±1db	3-5k±1db	3-5000	3-10,000	3-5,000	3-5,000	30-700	3-10,000	0.5-250	0.1-500	0.1-200
±3dB Hz	1-30,000	1-40,000	1-30,000	1-11,000	1-12,000	1-10,000	0.5-7,000	1-10,000	1-10,000	1-10,000	1-12,000	2-7,000	2-7,000	2-11,000	1-15,000	1-10,000	2-11,000	10-2,000	1-11,000	0.2-1000	0.03-1200	0.03-300
Turn-On-Time (2 % of final bias) Seconds	5	5	5	5	5	5	5	2	2	2	2	0.3	0.3	0.3	N/A	0.3	0.2	0.2	1.5	1.5	1.5	1.5
Amplitude Range (± 1%) ±g(pk)	550	500	500	500	50	50	50	80	80	80	80	80	500	65	–	70	55	550	60	8	3	.7
Resonant Frequency KHz	80	90	60	50	25	25	20	18	18	20	25	18	18	22	60	25	22	22	25	2.3	1.7	0.6
Noise																						
Broadband 2.5 Hz to 25 KHz µ g (rms)	1500	1500	1500	1500	250	200	200	250	250	250	250	300	3000	300	–	250	25	1500	250	40	50	20
Spectral 10 Hz µ g√Hz	100	100	100	100	14	14	14	12	12	12	12	28	280	26	–	12	8	200	12	4.0	1.0	0.6
100 Hz µ g√Hz	25	25	25	25	4	4	4	5	5	5	5	8	80	8	–	5	4	40	5	2.0	0.2	0.2
1000 Hz µ g√Hz	10	10	10	10	3	3	3	2	2	2	2	4	40	4	–	2	2	15	2	1.5	0.08	0.06
Transverse Sensitivity (typical)	<5%	<5%	<5%	<5%	<5%	<5%	<5%	<7%	<7%	<5%	<5%	<7%	<7%	<5%	<5%	<5%	<5%	<5%	<5%	<5%	<5%	<5%
Output Impedance OHM	1000	1000	1000	1000	1000	1000	1000	100	100	100	1000	1000	1000	1000	–	1000	1000	1000	1000	1000	1000	1000
EMI/RFI Protection dB	20	52	52/20	52	52	52	52	52	52	52	80	52	52	52	52	80	52	52	80	52	52	52
ESD Protection	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes
Ground Isolation M ohm	–	100	100/–	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100
Bias Volts (nom.)	7	7	7	7	7	7	7	12	12	12	12	12	12	12	–	10	8	10	10	7	7	7
Power Vdc, 2-4ma	15-30	15-30	15-30	15-30	15-30	15-30	15-30	24-30	24-30	24-30	24-30	24-30	24-30	24-30	–	24-30	20-30	24-30	24-30	15-30	15-30	15-30
Temperature Operating (°C/°F)																						
Minimum	-40/-40	-40/-40	-40/-40	-40/-40	-40/-40	-40/-40	-40/-40	-20/-10	-20/-10	-40/-40	-40/-40	-20/-10	-20/-10	-40/-40	-40/-40	-40/-40	-40/-40	-40/-40	-40/-40	-40/-40	-40/-40	-40/-40
Maximum	121/250	121/250	121/250	121/250	121/250	121/250	121/250	95/200	95/200	121/250	121/250	95/200	95/200	121/250	180/350	180/350	180/350	180/350	121/250	121/250	121/250	121/250
Size height (in/cm)	.30/.8	.45/1.1	.86/2.2	1.1/2.7	1.2/3.0	1.2/3.0	1.2/3.0	1.5/3.8	2.3/5.8	1.5x1.7x1.7	2.0/5.1	.85/2.1	.85/2.1	.85/2.1	1.1/2.8	2.4/6.1	.85/2.1	.85/2.1	1.7/4.3	3/7.6	2.4x2.2x2.2	2.5/6.4
flats (in/cm)	.25/.6	.3/.7	.5/1.3	1.0/2.5	.75/1.9	.75/1.9	.75/1.9	1.0.2.5	1.0/2.5	3.8x4.3x4.3	.9/2.2	1.0/2.5	1.0/2.5	1.0/2.5	.6/1.5	.9/2.2	1.0/2.5	1.0/2.5	.8/2.0	1.3/3.3	6.1x5.6x5.6	2/5.1
Weight grams (without cable)	.7	2	16	10	35	50	23	55	90	189	90	30	30	30	28	110	30	30	60	252	1250	430
Seal	epoxy	epoxy	epoxy	epoxy	epoxy	epoxy	epoxy	hermetic	hermetic	hermetic	welded	welded	welded	welded	epoxy	welded	welded	welded	welded	welded	hermetic	epoxy
Connector (S=Side, T=Top)	S, cable to 10-32	S, cable to 10-32	S, 10-32	S, 4-Pin	S, 10-32	S, cable to 1/4-28	S, 10-32	S, cable to BNC	T, cable to BNC	T, 4-Pin	T, 2-Pin	S, cable to BNC	S, cable to BNC	S, cable to BNC	S, 10-32	T, 2-Pin	S, cable to BNC	S, cable to BNC	S, cable 15 to PT	T, 2-Pin	S, 2-Pin	S, Coax
Mounting Stud	None	6-32	10-32	10-32	1/4-28	1/4-28	10-32	1/4-28	1/4-28	1/4-28	1/4-28	10-32	10-32	10-32	10-32	1/4-28	10-32	10-32	1/4-28	1/4-28	2x10-32	1/4-28
Design	iso. comprs	iso. comprs	iso. comprs	iso. comprs	iso. comprs	iso. comprs	iso. comprs	iso. comprs	iso. comprs	iso. comprs	iso. comprs	shear	shear	shear	iso. comprs	iso. comprs	shear	shear	iso. comprs	shear	shear	diaphragm
Sensor Highway® Compatible	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes	yes
Other Versions Available	• 9002A Low Noise • 100 mv/g	• Case Gnd • 50 KHz • Shock		• 3000X 7 KHz				• FM Intrinsically safe • Low Bias			• FM Intrinsically safe			• FM Intrinsically safe • TUFKEM Cable			• TUFKEM Cable	• TUFKEM Cable		• 5100 w/TNC		

1. As a part of continual product improvement, specifications subject to change without notice.
2. Registered trademarks of Vibra-Metrics, Inc. include: Sensor Highway, IsoShield, TUFKEM.
3. Calibration supplied with accelerometer is 10-10 KHz.

** Available on selected models. Consult factory for details.

Bulletin No. VM172

Vibra·Metrics®, Inc.
1014 Sherman Avenue • Hamden, CT 06514 • Tel (203) 288-6158 • Fax (203) 288-4937

LIFETIME WARRANTY ON 8000 SERIES**

* Patent No. 5,473,941

FOR APPLICATIONS ASSISTANCE OR ORDER PLACEMENT CALL OUR HOT LINE 1-800 U-SENS-IT

Technical Associates Level I
© Copyright 2001 Technical Associates of Charlotte, P.C.

VIBRA•METRICS ACCELEROMETER SELECTOR GUIDE

GENERAL AREA OF APPLICATION

Lab/ Lab Use — Ideal where small size, high sensitivity and broad frequency response are required for accurate structural testing.

General Purpose — Laboratory Shake Testing, Acceleration, Velocity, Displacement Control, Production Screening, useful for most vibration measurements.

MODEL NUMBER	9001A	1000	1001	2006	TRIAX 3000	TRIAX 3130	1018	1019	1020S
SPECIAL FEATURES									
Low Impedance for Long Lines	●	●	●	●	●	●	●	●	●
Physical Shock Protection	●	●	●	●	●	●	●	●	●
Electric Overload Protection	●	●	●	●	●	●	●	●	●
Electrically Isolated	—	●	—	●	●	●	●	—	●
Internal Faraday Shielding	—	—	—	●	—	—	—	—	—
Glass Fused Connector	—	—	—	—	—	—	—	—	—
Removable Stud	—	—	—	●	●	●	—	●	—
Ultrasonic Range (above 20KHz)	●	●	●	●	—	—	●	●	—
Very Low Noise	●	●	●	●	●	●	●	●	●
SPECIAL OPTIONS									
50 Ohm Output	●	●	●	●	●	●	●	●	●
±2% Sensitivity version	—	●	●	●	●	●	●	●	●
Magnetic Mount available	—	●	●	●	●	—	●	●	●
Low Frequency 0.5 Hz version	—	●	●	●	●	—	●	●	●
No Stud (for epoxy mounting)	Std.	●	●	—	—	—	—	—	—
Splashproof	●	●	●	—	—	●	●	●	●
Pre-filter	—	—	—	—	—	—	—	—	●
OPERATING SPECIFICATIONS									
Sensitivity mv/g @ 100 Hz ±5%	8-9	10	10	10	10	10	10	10	100
Frequency Range Hz ±5%	3 - 18K	3 - 10K	3 - 10K	3 - 10K	3 - 10K	3 - 10K	3 - 10K	3 - 10K	3 - 10K
Amplitude Linearity ±2% (g)	600	480	480	480	480	480	480	480	48
Max. Accel w/o Clipping (±g/VDC) Pk.	750 15	500 15	500 15	500 15	500 15	500 15	500 15	500 15	50 15
Max. Shock w/o Damage (g)	10000	10000	10000	10000	5000	5000	10000	10000	5000
Resonant Frequency Nom. (KHz)	80	90	90	60	50	50	60	60	25
Noise, g (rms), Wide Band	.0012	.0012	.0012	.0012	.0015	.0015	.0012	.0012	.0015
Transverse Sens (%) Max.	5	5	5	5	5	5	5	5	5
Sens. Temperature (%) (Nominal-Static Soak)									
−65°F to +100°F 38°C	+5 −5 −8	+5 −5 −8	+5 −5 −8	+5 −5 −8	+5 −5 −8	+5 −5 −8	+5 −5 −8	+5 −5 −8	+5 −5 −8
100°F to 150°F 66°C									
100°F to 250°F 120°C									
Thermal Shock g/°C Nominal	4	4	4	4	4	3	2	2	0.5
Base Strain g/micro Strain, nom.	.005	.005	.005	.005	.0015	.003	.003	.003	.008
Output Impedance - Ohms	1000	1000	1000	1000	1000	1000	1000	1000	1000
Part Number	9353031	9353018	9353027	9354533	9354323	9354707	9353173	9353174	9353205
PHYSICAL SPECIFICATIONS									
Weight - Grams (Nominal)	0.75	2	1.9	6	10	7	16	16	35
Size - Height in/cm	0.32 .81	0.44 1.12	0.44 1.12	0.72 1.83	1.08 2.74	0.61 1.55	0.86 2.18	0.82 2.08	1.20 3.05
Flats in/cm	0.25 0.63	0.28 0.71	0.28 0.71	0.438 1.11	0.98 2.48	0.25 0.63	0.50 1.27	0.50 1.27	0.75 1.91
Wiring	Coax	Coax	Coax	Coax	4 Wire Shielded	3 Coax	Coax	Coax	Coax
Power - Voltage	15 - 30	15 - 30	15 - 30	15 - 30	15 - 30	15 - 30	15 - 30	15 - 30	15 - 30
Current (ma)	1 - 4	1 - 4	1 - 4	1 - 4	1 - 4 Axis	1 - 4 Axis	1 - 4	1 - 4	1 - 4
Grounding	Case Ground	Isolated	Case Ground	IsoShield	Isolated	Isolated	Isolated	Case Ground	Isolated
Connector - Location	6' integral cable side	6' integral cable side	6' integral cable side	top	side	integral side	side	side	side
Type	10 - 32	10 - 32	10 - 32	10 - 32	4 - Pin	10 - 32	10 - 32	10 - 32	10 - 32
Mounting Stud	None	6 - 32	6 - 32	10 - 32	10 - 32	10 - 32	10 - 32	10 - 32	1 4 - 28
Material - Base	Titanium	Titanium	Titanium	Titanium	Titanium	303 SS	303 SS	303 SS	Titanium
Material - Case	Titanium	Titanium	Titanium	Titanium	Titanium	Titanium	303 SS	303 SS	Titanium
Seal	epoxy	epoxy	epoxy	epoxy	epoxy	epoxy	epoxy	epoxy	epoxy

P71291

Calibrations are traceable to N.I.S.T. to 10KHz and conform to MIL-STD-45662A and MIL-I-45208A.

SPECIAL OEM DESIGNS AVAILABLE

© Copyright 2001 Technical Associates of Charlotte, P.C.

(203)-288-6158
1-800-U-SENS-IT
FAX (203)-288-4937

1014 SHERMAN AVENUE Vibra• Metrics® INC. HAMDEN, CONNECTICUT 06514

MODEL 5100 ACCELEROMETER

LOW FREQUENCY - HIGH OUTPUT - SHEAR DESIGN

Industrial accelerometer designed for industrial mills and plants with low RPM machinery that require low frequency and high sensitivity specifications. The 5100 features rugged top connector, internal isolation, splash-proof construction along with the ability to measure 5 RPM and 10μg vibrations. Internal electronics will allow cable runs up to 1000 feet. Shear Design virtually eliminates base strain, thermal transients, and other phenomenon that are accentuated at low frequencies.

Operating Specifications

Sensitivity ±5% @ 100Hz	500 mv/g
Frequency Range ±5%	0.5 Hz to 500 Hz
±3 db	0.1 Hz to 700 Hz
Amplitude Range	±10 g
Transverse Sensitivity	7% maximum
Case Noise Rejection	-50 db
Maximum Shock without damage	5,000 g
Mounted Resonant Frequency (nom.)	2.3 KHz
Bias Voltage (nom.)	7 v
Output Impedance	1,000Ω
Isolation	Internally Isolated

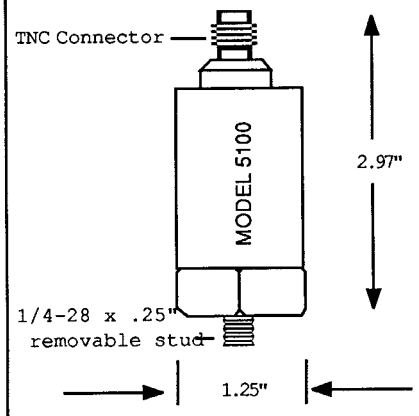
Operating Specifications (cont'd.)

Noise Floor (rms)	20 microvolts
Power Requirements	15-30 vdc, 1-4 ma constant current

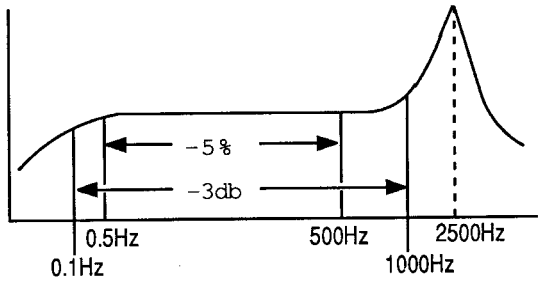
Physical Specifications

Size (inches)	2.97" high x 1.25" dia
Weight	9.5 oz.
Connector	Top TNC
Material	303 Stainless Steel
Mounting	1/4 - 28 Stud (removeable)
Weatherseal Rating	Splash-Proof
Temperature	-65° to +250°F

Model 5100
Outline Drawing



Model 5100
Typical Frequency Response

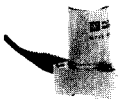
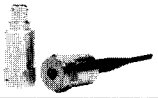
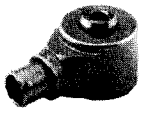


© Copyright 2001 Technical Associates of Charlotte, P.C.

Internally Amplified Vibration Sensors

Industrial, General Purpose Accelerometers

These rugged, general purpose sensors are ideal for machinery monitoring and predictive maintenance applications. They are used on machinery monitoring systems worldwide.

	Sensitivity, mV/g	Acceleration Range, g	Frequency Response, Hz ±5%, ±10%	Temperature Range, °C	Weight, grams	Case Material	Mounting
756 	100	.0007 - 50	3 - 1,000 2 - 5,000	-50 to 90	180	stainless steel	three #8 x ⅞" screws
762	10	.0015 - 500	3 - 10,000 2 - 12,000	-50 to 120	80	stainless steel	¼-28 tapped hole
766	100	.00055 - 50	3 - 10,000 2 - 12,000	-50 to 120	80	stainless steel	¼-28 tapped hole
768* 	500	.00002 - 10	3 - 4,000 2 - 6,000	-50 to 120	90	stainless steel	¼-28 tapped hole
776	100	.0002 - 40	3 - 2,500 2 - 5,000	0 to 85	32	stainless steel	¼-28 tapped hole
793	100	.0007 - 50	3 - 5,000 2 - 7,000	-50 to 120	110	stainless steel	¼-28 tapped hole
793L*	500	.00001 - 10	0.6 - 700 0.4 - 1,000	-50 to 120	135	stainless steel	¼-28 tapped hole
793LS	500	.00001 - 10	0.6 - 700 0.4 - 1,000	-50 to 120	135	stainless steel	¼-28 tapped hole
793S	100	.0007 - 50	3 - 5,000 2 - 7,000	-50 to 120	110	stainless steel	¼-28 tapped hole
796 	100	.0008 - 50	3 - 5,000 2 - 7,000	-50 to 120	100	stainless steel	¼-28 captive screw

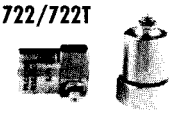
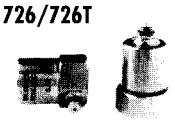
*** Ultra low noise sensor.**

© Copyright 2001 Technical Associates of Charlotte, P.C.

Internally Amplified Vibration Sensors

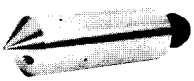
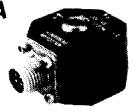
Lightweight, Compact Sensors

Lightweight, compact sensors are used in many laboratory research and light industrial applications. Their low mass makes them effective in applications where transducer mass must be minimized and high signal to noise ratio must be maintained.

	Sensitivity, mV/g	Acceleration Range, g	Frequency Response, Hz ±5%, ±10%	Temperature Range, °C	Weight, grams	Case Material	Mounting
722/722T 	10	.0002 - 500	3 - 10,000 2 - 12,000	-50 to 120	34	stainless steel	10-32 tapped hole
726/726T 	100	.0001 - 50	3 - 10,000 2 - 12,000	-50 to 120	34	stainless steel	10-32 tapped hole
728/728T*	500	.00003 - 10	3 - 4,000 2 - 6,000	-50 to 120	44	stainless steel	10-32 tapped hole
732A/732T	10	.0005 - 250	3 - 15,000 1 - 25,000	-50 to 120	15	stainless steel	10-32 tapped hole
736/736T	100	.00015 - 50	6 - 15,000 2 - 25,000	-50 to 120	15	stainless steel	10-32 tapped hole

Triaxial Accelerometers

Triaxial units contain three mutually perpendicular accelerometers in a compact housing, allowing the evaluation of three dimensional motion.

	Sensitivity, mV/g	Acceleration Range, g	Frequency Response, Hz all channels±10%	Temperature Range, °C	Weight, grams	Case Material	Mounting
733	10	.00015 - 250	2 - 4,000	-50 to 120	23	hardcoated aluminum	10-32 captive screw
755G* 	3	.000005 - 1.5	6 - 4000	-50 to 80	1900	titanium	——
993A 	100	.0002 - 50	2 - 2,000	-50 to 120	90	hardcoated aluminum	¼ - 28 captive screw

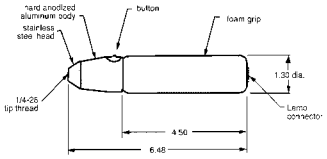
*** Ultra low noise sensor.**

© Copyright 2001 Technical Associates of Charlotte, P.C.

Model HA8
Hand-Held Vibration Probe - Acceleration Output

Features:
- Rugged
- Easy to use
- Shock protected with intrinsic overload protection
- Built-in electronics
- Miswiring protection

Applications:
- Predictive maintenance
- Bearing analysis
- Ship board machinery
- Heavy machinery
- Compressors



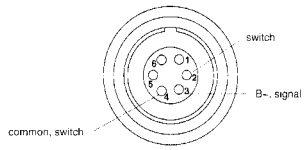
Model HA8

SPECIFICATIONS	HA8	UNITS
DYNAMIC		
Sensitivity, ±5 %, 25 °C	100	mV/g
Acceleration Range	50	g peak
Amplitude Nonlinearity	1	%
Frequency Response ±10 %:		
screwed down using mounting stud	2 - 2000	Hz
through probe tip	5 - 500	Hz
ELECTRICAL		
Power Requirement, voltage source	18 - 30	VDC
current regulating diode	2 - 10	mA
Electrical Noise, equiv. g, nom.:		
Broadband, 2.5 Hz to 25 kHz	100	µg
Spectral 10 Hz	10	µg/√Hz
100 Hz	5	µg/√Hz
1000 Hz	3	µg/√Hz
Output Impedance, max.	100	Ω
Bias Output Voltage, nom.	10	VDC
Grounding	case isolated, internally shielded	
ENVIRONMENTAL		
Vibration Limit	500	g peak
Shock Limit	100	g peak
Electromagnetic Sensitivity, equiv. g.		µg/gauss
PHYSICAL		
Weight	150	grams
Case Material	stainless steel head, hard anodized aluminum body	
Grip Material	black neoprene foam	
Output Connector (socket)	Lemo EGG1B306CNL	
Mating Connector (pins)	Lemo FGG1B306CNADA62 (R6LM)	

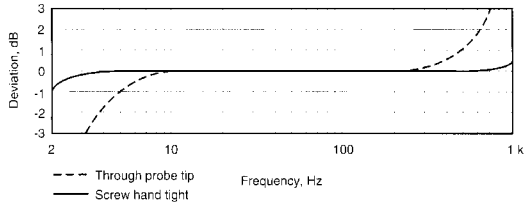
OPTIONS: Customer specified sensitivity, connector.

ACCESSORIES AVAILABLE: Cabling, power supplies, amplifiers, signal conditioners, probe tips.

Connector Pinout



TYPICAL FREQUENCY RESPONSE



Due to continued research and product development, Wilcoxon Research reserves the right to amend this specification without notice.

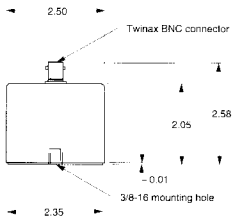
Wilcoxon Research, 2096 Gaither Road, Rockville, Maryland 20850 USA • 1-800-VIB-SENS or (301) 330-8811 • Fax: (301) 330-8873

© Copyright 2001 Technical Associates of Charlotte, P.C.

Model 731
Ultra-Quiet Seismic Accelerometer

Features:
- Ultra high sensitivity
- Ultra low-noise electronics for clear signals at very low vibration levels
- Low frequency capable
- Built-in electronics
- Filtered to eliminate high frequencies
- Capable of measuring to sub micro-g levels
- Miswiring protection

Applications:
- Foundations
- Earth tremors
- Structures
- Microscopes
- Optical equipment
- Laboratory research
- Semiconductor manufacturing



Model 731

SPECIFICATIONS	731	UNITS
DYNAMIC		
Sensitivity, ±5 %, 25 °C	10	V/g
Acceleration Range	0.5	g peak
Amplitude Nonlinearity	1	%
Frequency Response:		
±5 %	0.15 - 200	Hz
±10 %	0.10 - 300	Hz
±3 dB	0.05 - 500	Hz
Resonance Frequency, mounted, nom.	950	Hz
Transverse Sensitivity, max.	1	% of axial
Temperature Response	see graph	
ELECTRICAL		
Power Requirement[1], voltage source	18 - 30	VDC
current regulating diode	2 - 10	mA
Electrical Noise, equiv. g, nom.:		
Broadband, 2.5 Hz to 25 kHz	0.2	µg
Spectral 2 Hz	0.03	µg/√Hz
10 Hz	0.01	µg/√Hz
100 Hz	0.004	µg/√Hz
Output Impedance, max.	100	Ω
Bias Output Voltage, nom.	9	VDC
Grounding	isolated	
ENVIRONMENTAL		
Temperature Range	-10 to 65	°C
Vibration Limit	10	g peak
Shock Limit[2]	fragile	
Electromagnetic Sensitivity, equiv. g.	20	µg/gauss
Base Strain Sensitivity	0.0001	g/µstrain
PHYSICAL		
Weight	540	grams
Case Material	anodized aluminum	
Mounting	⅜ - 16 tapped hole	
Output Connector	Amphenol 31-2225	
Cabling: Mating Connector	Amphenol 31-224 (R2T)	
Standard Cable	J9, 2 conductor shielded, PVC jacket, 10 pF/ft.	

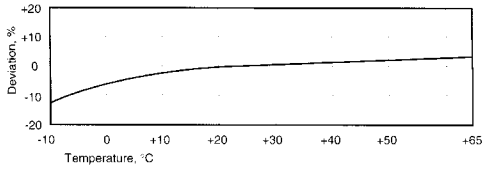
NOTES: [1] Model P31 powering unit is recommended for single channel battery operation. Model PR710-1 is available for up to 10 channels, rackmount, line powered applications; consult the factory.
[2] Will withstand a six inch drop to concrete.

OPTIONS: Customer specified sensitivity or frequency response, standard BNC connector.

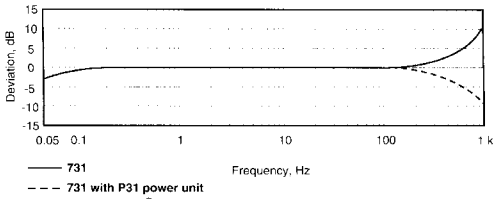
ACCESSORIES SUPPLIED: SF7 mounting stud, R2T2T-J9-10 cable assembly, panel mount receptacle (when sold without power unit).

ACCESSORIES AVAILABLE: Model P31 and PR710-1 powering units, magnetic mounting bases, cementing studs, triaxial mounting block.

TYPICAL ACCELEROMETER TEMPERATURE RESPONSE



TYPICAL FREQUENCY RESPONSE



Due to continued research and product development, Wilcoxon Research reserves the right to amend this specification without notice.

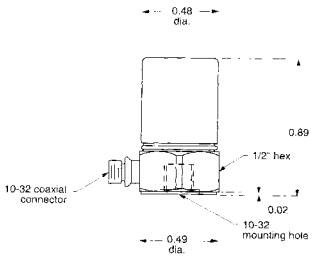
Wilcoxon Research, 2096 Gaither Road, Rockville, Maryland 20850 USA • 1-800-VIB-SENS or (301) 330-8811 • Fax: (301) 330-8873

© Copyright 2001 Technical Associates of Charlotte, P.C.

Model 732B
High Frequency Accelerometer

Features:
- Rugged
- Versatile
- Wide frequency range
- Standardized sensitivity
- High resonance frequency
- High frequency range
- Hermetically sealed

Applications:
- Pumps
- Turbines
- Helicopters
- Ship board machinery
- Laboratory research
- Compressors
- High frequency testing



Model 732B

SPECIFICATIONS	732B	UNITS
DYNAMIC		
Sensitivity, ±5 %, 25 °C	10	mV/g
Acceleration Range	250	g peak
Amplitude Nonlinearity	1	%
Frequency Response:		
±5 %	3 - 15,000	Hz
±3 dB	1 - 25,000	Hz
Resonance Frequency, mounted, nom.	50	kHz
Transverse Sensitivity, max.	5	% of axial
Temperature Response	see graph	
ELECTRICAL		
Power Requirement, voltage source	18 - 30	VDC
current regulating diode[1]	0.5 - 10	mA
Electrical Noise, equiv. g, nom.:		
Broadband, 2.5 Hz to 25 kHz	2000	µg
Spectral 10 Hz	100	µg/√Hz
100 Hz	30	µg/√Hz
1,000 Hz	10	µg/√Hz
10,000 Hz	5	µg/√Hz
Output Impedance, max.	800	Ω
Bias Output Voltage, nom.[2]	6	VDC
Grounding	case grounded	
ENVIRONMENTAL		
Temperature Range	-50 to 120	°C
Vibration Limit	500	g peak
Shock Limit	5000	g peak
Electromagnetic Sensitivity, equiv. g.	100	µg/gauss
Base Strain Sensitivity	0.005	g/µstrain
PHYSICAL		
Weight	15	grams
Case Material	stainless steel	
Mounting	10-32 tapped hole	
Output Connector	10-32 coaxial	
Cabling: Mating Connector	Microdot 10-32 (R1)	
Standard Cable	J93, coaxial, Teflon jacket, 30 pF/ft	

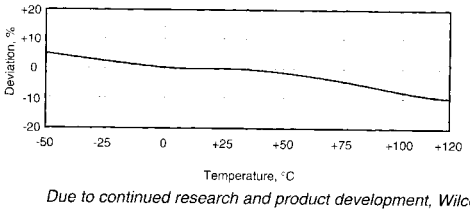
NOTES: [1] A maximum current of 6 mA is recommended for operating temperatures in excess of 100 °C.
[2] Bias output voltage will vary between 5-10 volts depending upon the value of the constant current diode used.

OPTIONS: Customer specified filtering and sensitivity, top connector (BNC), PVC jacketed cable.

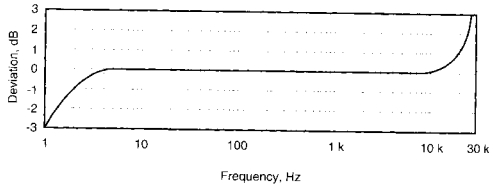
ACCESSORIES SUPPLIED: SF1 mounting stud, R12-J93-10 cable assembly.

ACCESSORIES AVAILABLE: Power supplies, amplifiers, signal conditioners, cementing studs, magnetic mounting bases, isolating studs.

TYPICAL TEMPERATURE RESPONSE



TYPICAL FREQUENCY RESPONSE

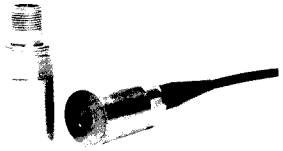


Due to continued research and product development, Wilcoxon Research reserves the right to amend this specification without notice.

Wilcoxon Research, 2096 Gaither Road, Rockville, Maryland 20850 USA • 1-800-VIB-SENS or (301) 330-8811 • Fax: (301) 330-8873

© Copyright 2001 Technical Associates of Charlotte, P.C.

Model 793
Industrial Piezoelectric Accelerometer

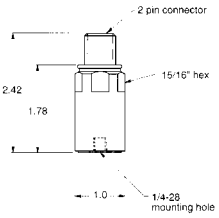


Features:

- Intrinsic safety certified option
- Rugged, general purpose
- Corrosion-resistant
- Priced economically for single and OEM quantities
- Hermetic seal
- Standardized sensitivity
- Ground isolated
- ESD protection
- Miswiring protection

Applications:

- Engine monitoring
- Machinery monitoring
- Ship board machinery
- Shock & vibration
- Predictive maintenance
- Shock pulse measurements



Model 793

SPECIFICATIONS	793	UNITS
DYNAMIC		
Sensitivity, ±5 %, 25°C	100	mV/g
Acceleration Range[1]	50	g peak
Amplitude Nonlinearity	1	%
Frequency Response:		
±5 %	3 - 5,000	Hz
±10 %	2 - 7,000	Hz
±3 dB	1 - 15,000	Hz
Resonance Frequency, mounted, nom.	26	kHz
Transverse Sensitivity, max.	5	% of axial
Temperature Response	see graph	
ELECTRICAL		
Power Requirement:		
voltage source	18 - 30	VDC
current regulating diode[1, 2]	2 - 10	mA
Electrical Noise, equiv. g, nom.:		
Broadband, 2.5 Hz to 25 kHz	700	µg
Spectral 10 Hz	10	µg/√Hz
100 Hz	5	µg/√Hz
1000 Hz	5	µg/√Hz
Output Impedance, max.	100	Ω
Bias Output Voltage, nom.	12	VDC
Grounding	case isolated, internally shielded	
ENVIRONMENTAL		
Temperature Range	-50 to 120	°C
Vibration Limit	500	g peak
Shock Limit	5000	g peak
Electromagnetic Sensitivity, equiv. g.	15	µg/gauss
Humidity Limit	100	% relative
Base Strain Sensitivity	0.004	g/µstrain
PHYSICAL		
Weight	110	grams
Case Material	stainless steel	
Mounting	¼ - 28 tapped hole	
Output Connector	MIL-C-5015 2-pin	
Cabling: Mating connector	Amphenol 97-3106A-10SL-4S (R6)	
Recommended Cable	J5, coaxial, polyurethane jacket, 30 pF/ft	

NOTES: [1] Signal distortion can occur when measuring high vibration levels, especially with long cables. A 24-30 VDC powering source is recommended for minimizing distortion. Current regulating diode must be at least 2 mA plus 1 mA per 1000 pF of cable capacitance.
[2] A maximum current of 6 mA is recommended for operating temperatures in excess of 100 °C.

Connections
Pin A ..... Signal, power
Pin B ..... Common

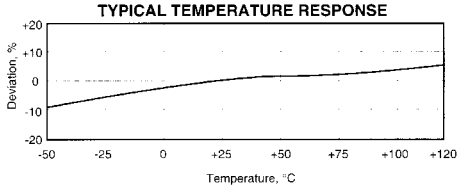
OPTIONS: Intrinsic safety "E" (793E) certified by Factory Mutual for use in the following environments: Classes I, II, III, Div. I, Groups A, B, C, D, E, F, G, 10 ft. sealed integral cable (see 793S/SG), temperature sensor "T", ultra low noise "U", customer specified filtering, sensitivity, cable length.

ACCESSORIES SUPPLIED: SF6 mounting stud.

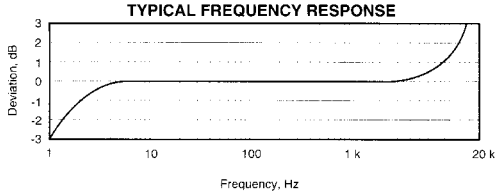
FM APPROVED

ACCESSORIES AVAILABLE: Magnetic mounting bases, cementing studs, cable assembly R62-J5-XX, power supplies, amplifiers, signal conditioners.

TYPICAL TEMPERATURE RESPONSE



TYPICAL FREQUENCY RESPONSE

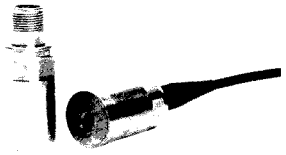


Due to continued research and product development, Wilcoxon Research reserves the right to amend this specification without notice.

Wilcoxon Research, 2096 Gaither Road Rockville, Maryland 20850 USA • 1-800-VIB-SENS or (301) 330-8811 • Fax: (301) 330-8873

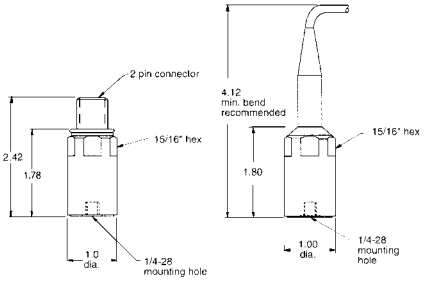
© Copyright 2001 Technical Associates of Charlotte, P.C.

Model 793L
Low Frequency Piezoelectric Accelerometer



Features:
- Intrinsic safety certified option
- High sensitivity
- Ultra low-noise electronics for clear signals at very low vibration levels
- Low frequency capable
- Filtered to eliminate high frequencies
- Hermetic seal
- ESD protection
- Miswiring protection

Applications:
- Machinery monitoring
- Rotating machinery
- Low vibration level measurements
- Laboratory research



Model 793L **Model 793LS**

SPECIFICATIONS	793L	UNITS
DYNAMIC		
Sensitivity, ±5 %, 25 °C	500	mV/g
Acceleration Range[1]	10	g peak
Amplitude Nonlinearity	1	%
Frequency Response, nom.:		
±5 %	0.6 - 700	Hz
±10 %	0.4 - 1000	Hz
±3 dB	0.2 - 2300	Hz
Resonance Frequency, mounted, nom.	15	kHz
Transverse Sensitivity, max.	7	% of axial
Temperature Response	see graph	
ELECTRICAL		
Power Requirement, voltage source	18 - 30	VDC
current regulating diode[1,2]	2 - 10	mA
Electrical Noise, equiv. g, nom.:		
Broadband, 2.5 Hz to 25 kHz	10	µg
Spectral 2 Hz	1.5	µg/√Hz
10 Hz	0.4	µg/√Hz
100 Hz	0.2	µg/√Hz
Output Impedance, max.	100	Ω
Bias Output Voltage, nom.	10	VDC
Grounding	case isolated, internally shielded	
ENVIRONMENTAL		
Temperature Range	-50 to 120	°C
Vibration Limit	250	g peak
Shock Limit	2500	g peak
Electromagnetic Sensitivity, equiv. g.	5	µg/gauss
Humidity Limit	100	% relative
Base Strain Sensitivity	0.001	g/µstrain
PHYSICAL		
Weight	135	grams
Case Material	stainless steel	
Mounting	¼-28 tapped hole	
Output Connector	MIL-C-5015 2-pin	
Cabling[3]: Mating Connector	Amphenol 97-3106A-10SL-4S (R6)	
Recommended Cable	J5, coaxial, polyurethane jacket, 30 pF/ft	

NOTES: [1] Signal distortion can occur when measuring high vibration levels, especially with long cables. A 24-30 VDC powering source is recommended for minimizing distortion.
[2] A maximum current of 6 mA is recommended for operating temperatures in excess of 100 °C.
[3] Cabling for "S" option model is J61, coaxial, polyurethane jacket, 30 pF/ft.

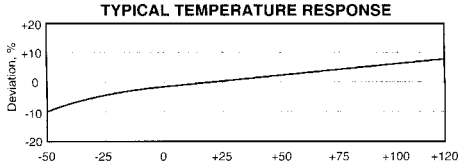
Connections
Pin A ..... Signal, power
Pin B ..... Common

OPTIONS: Intrinsic safety "E" (793LE) certified by Factory Mutual for use in the following environments: Classes I, II, III, Div. 1, Groups A, B, C, D, E, F, G, 10 ft. sealed integral cable "S", temperature sensor "T", customer specified sensitivity, filtering, cable length.

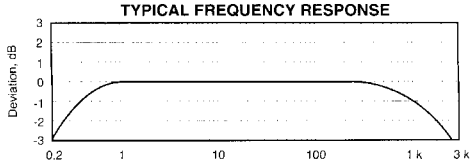
ACCESSORIES SUPPLIED: SF6 mounting stud.

ACCESSORIES AVAILABLE: R62-J5-XX cable assembly, magnetic mounting bases, cementing studs, power supplies, amplifiers, signal conditioners.

TYPICAL TEMPERATURE RESPONSE



TYPICAL FREQUENCY RESPONSE



Due to continued research and product development, Wilcoxon Research reserves the right to amend this specification without notice.

Wilcoxon Research, 2096 Gaither Road, Rockville, Maryland 20850 USA • 1-800-VIB-SENS or (301) 330-8811 • Fax: (301) 330-8873

© Copyright 2001 Technical Associates of Charlotte, P.C.

CHAPTER 5

ROLE OF SPIKE ENERGY, HFD AND SHOCK PULSE (SPM) & SPECIFICATION OF THEIR ALARM LEVELS AT VARIOUS SPEEDS

A. Spike Energy and Shock Pulse:

Ultrasonic instruments are designed to measure energy levels in roughly the 25,000 Hz to 100,000 Hz frequency range (1,500,000 - 6,000,000 CPM). These include Spike Energy[1] and Shock Pulse[2]. Each of these two parameters were developed in the 1970's and are designed to mechanically and electrically respond only to high frequency problem sources which excite the resonant frequency of their mounted transducer. Note the effective range for Spike Energy is from 5,000 Hz to 60,000 Hz. as shown in Illustration A. In the case of spike energy, the predominant accelerometer now in use (IRD 970) has a mounted natural frequency of about 30,000 Hz. Similarly, the reference mass within the shock pulse transducer is designed to respond at its resonant frequency of approximately 32,000 Hz.

Each of these technologies has proven worthwhile and capable of acting as effective tools at picking up initial stages of bearing wear if the user follows very strict measurement rules (these are covered in a Vibration Institute article entitled "The Use of Spike Energy for Fault Analysis and Machine Condition Monitoring" written by Joseph M. Shea and James K. Taylor of IRD). When bearings first begin to wear, they begin to excite natural frequencies of their components themselves (rolling elements, bearing races and cage) as discussed in the special rolling element bearing section in Chapter 6.07. One set of their natural frequencies is concentrated within the range of 500 to 2000 Hz (30,000 to 120,000 CPM). Another set is found within ultrasonic frequency ranges near the SPM and spike energy transducer natural frequencies. Therefore, when incipient wear just begins within rolling element bearings, the bearing components begin to impact exciting these natural frequencies which likewise excite the SPM and spike energy transducer natural frequencies. Basic defect mechanisms which generate ultrasonic response include:

a. Micro spalls and cracks from fatigue or overstressing caused by brinnelling, false brinnelling, misalignment, overload, incorrect sealing and improper fits.

b. Surface roughening from lack of sufficient lubrication.

c. Surface indentations from hard contaminant micropitting from electric current passing through the bearing.

In the case of spike energy, accelerometers have been designed which have mounted natural frequencies on the order of 30,000 Hz (1,800,000 CPM) in the ultrasonic range. The sharp pulses and broadband random ultrasonic excitation from the bearings will excite the accelerometer natural frequency whether the accelerometer is stud-mounted, magnet-mounted, or probe-mounted (obviously, the stud-mount gives the highest spike energy magnitudes and most repeatable results). But all three mountings will provide spike energy measurements.This is somewhat surprising when one recalls what happens to vibration measurement quality results when he goes from stud to magnet to probe mount. However, like the empirical data that has been captured on rolling element bearings showing that their set of ultrasonic natural frequencies shows little change whether "free-free" or with the bearings mounted in the machine, the same

© Copyright 2001 Technical Associates of Charlotte, P.C.

appears to be true in the case of spike energy (Ref.6). In “layman’s” terms, spike energy is generated when the incoming ultrasonic frequencies excite the accelerometer’s natural frequency, an electrical voltage response occurs which is then electronically conditioned and filtered, converting the end product to what might be called “impact energy” which is proportional to the incoming impact velocity. It is probably best thought of as a friction measuring parameter.

Spike energy pulses acting at high frequencies propagate rapidly through the structure. In the case of bearings, Reference 4 reports these elastic wave energy pulses travel through the bearing and surrounding structure at velocities approaching 16,000 ft/sec which is the velocity of sound (elastic) waves in steel (10,900 mi/hr!). When these impact waves encounter an interface of two materials or components, they lose much of their energy. This interface may be a region between the bearing outer race and its housing, or between one machine part and another to which this part is fastened. When these ultrasonic waves encounter an interface, some of the energy is transmitted through, and some is reflected back towards the source. The amount of energy reflected depends on a number of factors: the sharpness of the pulse, the difference in metals at the interface, how tightly the sources mate together, etc. In general, it is estimated that typically approximately 60% to 80% of the energy is reflected at each interface. **THEREFORE, THIS EXPLAINS WHY IT IS SO IMPORTANT TO KEEP THE MEASURING ULTRASONIC TRANSDUCER AS CLOSE AS POSSIBLE TO THE BEARING BEING EVALUATED.**

While this energy loss may be considered as a weakness, it also has an advantage. For example, when a bearing begins to fail, much of its vibration not only will exist within this bearing, but will transmit into other structures and bearings. On the other hand, ultrasonic energy is much more localized, making it easier to isolate the specific bearing having the problem.

Here, it is important to point out that not only does spike energy and shock pulse respond to bearing wear, but also such **ultrasonic measurements respond to each of the following**:

- bearing wear (as stated before)
- bearing lubrication
- cavitation
- rotor or seal rub
- belt squeal
- gear meshing
- sheave rub against a guard
- impact excitation of machine parts (i.e., valve in a reciprocating machine)
- steam and high pressure air flow

Since they do respond to all of these problems, this makes ultrasonic measurements of even greater use. In some cases, it might be confusing whether the problem is radiating from a bearing or from one of the other many sources listed above. However, if one remembers that ultrasonic energy dissipates rapidly (is very localized), it can help him in diagnosing which of several problems might exist. For example, in the case of a pump, if spike energy is high on both pump bearings, it is very likely that the pump might be experiencing cavitation or a seal rub. If the operator likewise records that he hears something like a “gravel sound”, it might make a strong case for cavitation. On the other hand, if on the same pump, spike energy were high only on one of the two bearings, it would suggest possible wear or lubrication problems with this one bearing.

© Copyright 2001 Technical Associates of Charlotte, P.C.

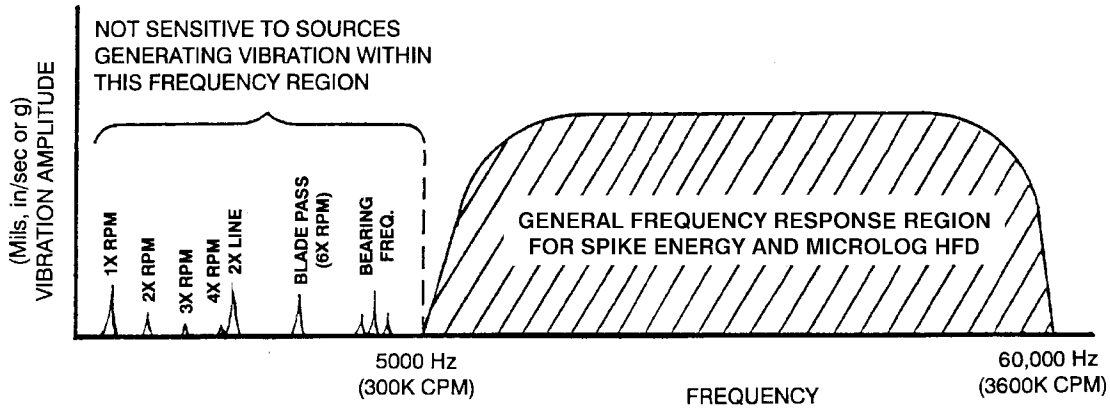


ILLUSTRATION A.
GENERAL FREQUENCY RESPONSE CURVE
FOR SPIKE ENERGY AND MICROLOG HFD

Finally, it is also important to point out **what ultrasonic measurements do not respond to**:

- unbalance
- misalignment
- bent shaft
- electrical problems
- eccentric rotors
- resonance
- structural looseness/weakness
- beat vibration problems

The reason that neither shock pulse nor spike energy respond to the above problems is that these problems tend to generate lower frequency vibration which is purposely filtered out in the design of the analyzer electronics [neither SPM nor spike energy are sensitive to problems which generate fault frequencies below 5000 Hz (300,000 CPM)]. Still, they are very useful in picking up the other problems such as bearing, lubrication, cavitation and rub problems.

Each of these technologies likewise have some disadvantages. In the case of shock pulse, the bearing bore and RPM (or bearing type and number) must be known in order to take its reference baseline (otherwise, the results are inaccurate). This is a disadvantage on many machines whose bearing model numbers and bores are not known. Also, since the shock pulse instrument usually makes only ultrasonic measurements, a second instrument is required to obtain vibration readings (particularly filtered vibration spectra) which will detect the lower frequency problems. In addition, since the shock pulse instrument is not yet included within any of the major predictive maintenance software offerings, its readings have to be manually entered into these PMP databases if one wants to both store and trend these measurements in his PMP software allowing him to directly compare vibration and shock pulse measurements in one trend graph. (However, SPM does now offer its own software for trending shock pulse readings themselves).

© Copyright 2001 Technical Associates of Charlotte, P.C.

In the case of spike energy, one of its disadvantages currently is that its amplitude response is highly dependent on which specific accelerometer (and mounting) is used to make the measurement. Amplitudes may differ by a factor of 5X to 10X from one accelerometer to another (however, if one always measures spike energy using the same accelerometer and mounting at identical locations from one survey to the next, meaningful trends and alarm levels can be established). One advantage spike energy has over SPM is that one does not have to know the bearing bore and model number to take a reading. However, another disadvantage has been found that when spike energy measurements are taken in the presence of steam, the steam itself can cause variant readings.

The advantages for both shock pulse and spike energy readings lie in the fact that they mandate one always take measurements at identical points from one survey to the next (not doing so will cause trends to vary widely). This will improve the accuracy, reliability and repeatability of the data from one survey to the next. In addition, if readings are taken with hand-held probes rather than magnet or stud mounted, simply a change in force on the transducer by the operator can raise or lower the subsequent reading. In both cases, if readings can be taken from stud-mounted or magnet mount locations, trends can greatly be improved. Spike energy measurements have also proven fairly repeatable using magnet mounts (particularly if the magnet exerts high force and can be mounted on a uniform surface), though not to the quality and repeatability of readings from stud mounts.

In any case, ultrasonic measurements are recommended on high-speed machines or lower speed machines having high frequency vibration sources (rolling element bearings, gears, etc). They can prove very effective in picking up many problems that might otherwise go unnoticed, particularly during incipient stages, but sometimes even in more advanced stages. In the case of bearings, each of these tools will pick up problems during the first of four definable stages of degradation (Ref. 9 & 10). Finally, meaningful severity charts are now being developed for each of these ultrasonic parameters. Figures 1 and 2 are included as example severity charts for spike energy and shock pulse, respectively.

High-Frequency Acceleration (HFD)

Some vendors have offered high-frequency acceleration measurements in addition to vibration with their predictive maintenance hardware and software systems. While this is similar to the ultrasonic measurements mentioned above, this is truly a different measurement parameter altogether. In fact, as the name implies, this is a banded high frequency acceleration measurement which measures the total energy existing between certain lower and higher cutoff frequencies. Most often, this high-frequency measurement is referred to as HFD and is marketed by some of the major predictive maintenance vendors (see "Endnotes " 3 & 4). Figure 2 shows severity levels for "Microlog HFD" as compared with those of spike energy and shock pulse. In general, note that Microlog HFD levels tended to be approximately 2 times those of spike energy (that is, about 100% higher) in this particular study.

© Copyright 2001 Technical Associates of Charlotte, P.C.

The thing which differentiates HFD measurements from those of shock pulse and spike energy is that HFD has not generally been an ultrasonic measurement, but instead a banded acceleration measurement. HFD by Vendor 3, is normally banded between approximately 5000 Hz and 20,000 Hz (300,000 - 1,200,000 CPM). However, this particular Vendor allows the user to specify lower and higher cutoff frequencies up to as high as 20,000 Hz. In these cases, it is understood that some users have specified their lower cutoff frequency as low as 1000 Hz up to 5000 Hz and their higher cutoff frequency anywhere from 3000 Hz to as high as 20,000 Hz. Microlog HFD, by Vendor 4, provides a fixed frequency range of 5000 Hz to 60,000 Hz (300,000 CPM - 3,600,000 CPM).

Like spike energy and shock pulse, HFD is sensitive to faults generating high frequencies such as bearing wear, gear wear, cavitation, etc. Due to its high frequency banding, HFD is not sensitive to problems such as unbalance, misalignment, eccentricity, etc.

ENDNOTES

1. “Spike Energy” was developed, patented and marketed by IRD Mechanalysis, Inc. based in Columbus, OH.

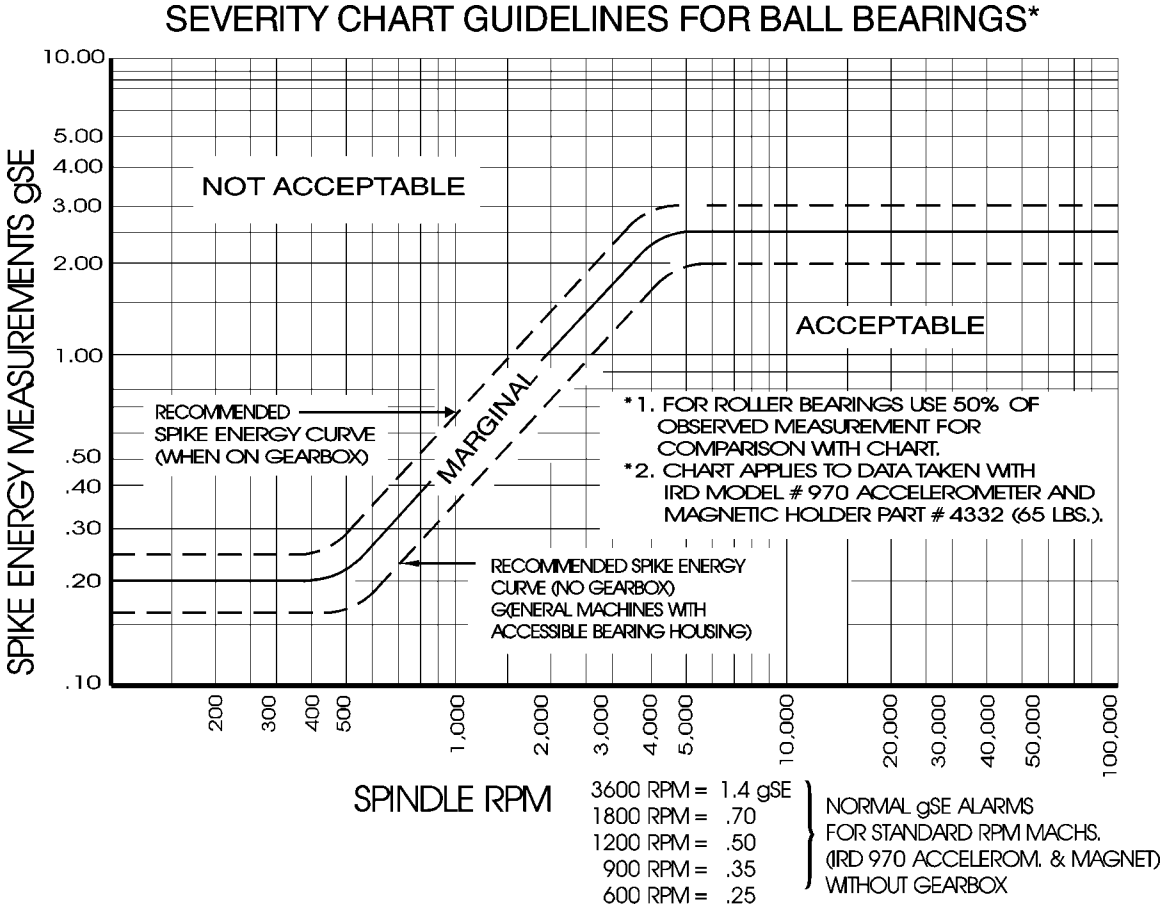
2. “Shock Pulse” was developed and marketed by SPM Instruments out of Marlborough, CT.

3. “HFD” was developed and marketed by CSI based in Knoxville, TN.

4. "Microlog HFD" was developed and marketed by SKF Condition Monitoring based in San Diego, CA.

© Copyright 2001 Technical Associates of Charlotte, P.C.

FIGURE 1
RECOMMENDED SPIKE ENERGY SEVERITY CHART (IRD SPIKE ENERGY)



SPIKE ENERGY™ MEASUREMENTS

Energy is generated by repetitive transient mechanical impacts. Such impacts typically occur as a result of surface flaws in rolling-element bearings or gear teeth.

This energy is conducted from its source through various paths to the outer surface of the machine structure, and is seen as a small-amplitude vibration at the surface. Accelerometers coupled to the surface generate corresponding electrical signal.

The accelerometer signals processed by unique filtering and detection circuitry to produce a single "figure of merit" related to the intensity of the original impacts. This figure of merit is expressed in "gSE" units.

SPIKE ENERGY™ gSE readings are measurements which can with experience, be correlated with the severity of the casual surface flaws. Even though gSE readings are affected by the nature of the conductive path between the impact source and the accelerometer, similar machine structures will provide a reasonable basis for comparison between the structures.

The gSE figure of merit has proven to be effective in detecting mechanical defects in meshing gears and rolling element bearings. The gSE measurement, when used in conjunction with conventional measurement of vibration velocity and acceleration, provides early indications of mechanical deterioration.

** When used with magnetic holders, accelerometers must be installed with a light coating of silicone grease and tightened to 40 in-lb. torque.

© Copyright 2001 Technical Associates of Charlotte, P.C.

FIGURE 2
MAINTENANCE DIAGNOSTICS VIBRATION AND HIGH FREQUENCY GENERAL TOLERANCE CHART FOR PROCESS MACHINERY WITH ROLLING ELEMENT BEARINGS①

Unfiltered Vibration Velocity	Shock Pulse & Acoustic Emission	Microlog HFD	IRD Spike Energy②	Severity Level and Alarm Status
Inches/Second Peak	Db	g	gSE	
1.5+	50+	5.0+	3.0+	Danger (Shutdown)
.75 - 1.49	40 - 49	3.0 - 4.99	1.50 - 2.99	Very Rough (Alert)
.40 - .74	30 - 39	1.50 - 2.99	.80 - 1.49	Rough (Alert)
.20 - .39	20 - 29	.75 - 1.49	.40 - .79	Fair (Acceptable)
.10 - .19	10 - 19	.30 - .74	.20 - .39	Good (Acceptable)
.01 - .09	1 - 9	.01 - .29	.01 - .19	Smooth (Acceptable)

① Provided by Charles Berggren of Monsanto
② Spike Energy Amplitudes measured using an IRD 970 accelerometer outfitted with IRD's 2-pole, 65 lb. magnet.

© Copyright 2001 Technical Associates of Charlotte, P.C.

CHAPTER 6

USE OF VIBRATION SIGNATURE ANALYSIS TO DIAGNOSE MACHINE PROBLEMS

6.0 USE OF VIBRATION SIGNATURE ANALYSIS

Probably the greatest shortcoming in today's predictive maintenance programs is the ability to diagnose the mechanical and electrical problems within the machine that are evidenced in the vibration signatures if the vibration exceeds preset overall and spectral alarm levels. For example, an ever increasing number of plants have PMP data collectors and software and successfully build large databases and capture vibration measurements at great numbers of points. However, surveys prove that less than 15% of such plants know how to properly set overall and spectral alarm levels. Then, even a smaller minority know how to diagnose the array of potential problems from the vibration spectra and related variables (i.e. spike energy) which are faithfully printed out by the PMP software for those points that exceed alarm. Therefore, the overriding purpose of this section is to begin to acquaint the reader on how to diagnose such problems from this data. Much valuable information is contained within these vibration spectra, but is only of use if the analyst can unlock its "secrets".

Therefore, Table 6.0 has been developed to put many of these "secrets" right at the fingertips of the analyst. Several hundred hours of research have gone into the development of this four page diagnostic chart. Please note that this chart not only provides text elaborating on vibration symptoms for various machine problems, but it also includes illustrations of "typical vibration spectra" for each problem covered. In addition, drawings are included to illustrate how phase reacts when such problems are predominant. Table 6.0 represents the best understanding to date of the author on how these problems are best diagnosed, based on approximately 16 years field experience in vibration signature analysis and research on a wide range of articles which have been written on the subject.

There are several key items included in Table 6.0. First, the plots under "TYPICAL SPECTRUM" column reveal invaluable information as to the source of the problem. When looking at such spectra, the analyst should ask questions similar to the following:

1. Which frequencies are present in spectrum and how do they relate to machine operating speed (that is, are the peaks present equal to 1X, 2X, 3X, 5.78X RPM or what)?

2. What are the amplitudes of each peak?

3. How do the frequency peaks relate to one another? (i.e., "2X RPM is much higher than 1X RPM"; "there is a large peak at 7.43X RPM"; "there are large number of operating speed harmonics present"; "there are high amplitude sidebands around gear mesh frequency"; "there are 7200 CPM sidebands around a large peak at 46X RPM"; etc.).

4. Finally, if there are significant amplitude peaks, what exactly is their source ("is 7.43X RPM a bearing defect frequency"; "is the 46X RPM peak equal to the number of rotor bars RPM"?).

© Copyright 2001 Technical Associates Of Charlotte, P.C.

As its column name implies, "TYPICAL SPECTRUM" is meant to be a representative signature for each problem listed in Table 6.0. These spectra are not intended to be all inclusive. For example, referring to "REMARKS" for the Angular Misalignment problem, note that while the typical spectrum for this problem shows high amplitude 1X RPM and 2X RPM peaks in the axial direction, the discussion shows that it is not unusual for either 1X, 2X or 3X RPM to dominate the whole spectrum when angular misalignment is the problem. In addition, it is not unusual for a machine to have two or more problems present at any one time. For example, if a machine simultaneously had both mechanical looseness and rotor unbalance, they each would contribute frequencies to its spectra which might show high 1X RPM in addition to multiple running speed harmonics.

The next column in Table 6.0 is entitled "PHASE RELATIONSHIP". Information on phase is provided for several of the problem sources listed. Amplitude reveals how much something is vibrating. Frequency relates how many cycles occur per unit of time. Phase completes the picture by showing just how the machine is vibrating. Of great importance, phase is a powerful tool in helping to differentiate which of several problem sources are dominant. For example, there are a large number of problems that generate vibration at 1X and 2X RPM. Using phase, one learns how the machine is vibrating, and in the process, helps zero in on just which problem is present. For example, Table 6.0 shows how phase reacts during the following scenarios:

1. **Force (or "static") unbalance** is evidenced by nearly identical phase in the radial direction on each bearing of a machine rotor.

2. **Couple unbalance** shows approximately a 180° out-of-phase relationship when comparing the outboard and inboard horizontal, or the outboard and inboard vertical direction phase on the same machine.

3. **Dynamic unbalance** is indicated when the phase difference is well removed from either 0° or 180°, but importantly is nearly the same in the horizontal and vertical directions. That is, the horizontal phase difference could be almost anything between the outboard and inboard bearings; but, the key point is that the vertical phase difference should then be almost identical to the horizontal phase difference (± 30°). For, example, if the horizontal phase difference between the outboard and inboard bearings is 60°, and the dominant problem is dynamic unbalance, the vertical phase difference between these two bearings should be about 60° (± 30°). If the horizontal phase difference varies greatly from the vertical phase difference, this strongly suggests the dominant problem is not unbalance.

4. **Angular misalignment** is indicated by approximately a 180° phase difference across the coupling, with measurements in the axial direction.

5. **Parallel misalignmen**t causes radial direction phase across the coupling to be approximately 180° out of phase with respect to one another.

6. **Bent shaft** causes axial phase on the same shaft of a machine to approach a 180° difference when comparing measurements on the outboard and inboard bearings of the same machine rotor.

7. **Resonance** is shown by a 90° phase change at the point when the forcing frequency coincides with a natural frequency, and approaches a full 180° phase change when the machine passes through the natural frequency (depending on the amount of damping present).
8. **Rotor rub** causes significant, instantaneous changes in phase.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

9. **Mechanical looseness/weakness due to base/frame problems or loose hold-down bolts** is indicated by nearly a 180° phase change when one moves his transducer from the machine foot down to its baseplate and then down to its support base.

10. **Mechanical looseness due to a cracked frame, loose bearing or loose rotor** causes phase to be unsteady with probable widely differing phase measurements from one measurement to the next. The phase measurement may noticeably differ every time you start up the machine, particularly if the rotor itself is loose and rotates on the shaft a few degrees with each startup.

Often, even though phase measurement capability is now offered by most data collectors, users do not use this powerful tool. If not used, this will severely limit the diagnostic capabilities of any program. However, currently it would be impractical to make phase measurements on all machinery during regular PMP surveys. Its greatest use comes into play when performing diagnostics on machines which have developed high vibration at 1X, 2X or 3X RPM, requiring investigation to detect the predominate cause(s) prior to taking corrective actions.

Note that "PHASE RELATIONSHIP" is illustrated in each of the first 8 problems of Table 6.0 since it is primarily with these problems that phase can be used to differentiate which problem(s) dominate. Phase is then discussed in many of the remarks for the remaining problems in Table 6.0, although it is not illustrated.

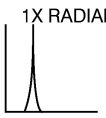
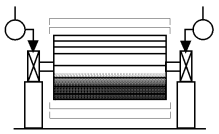
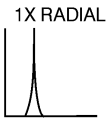
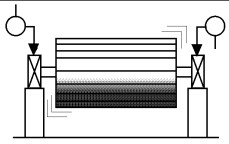
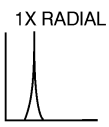
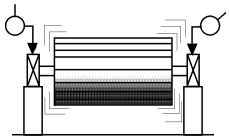
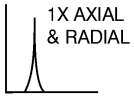
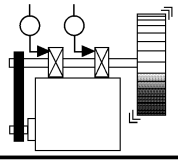
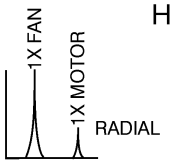
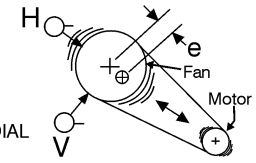
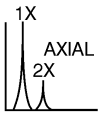
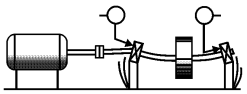
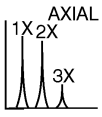
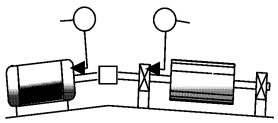
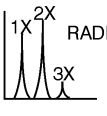
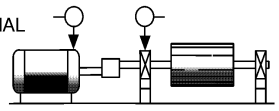
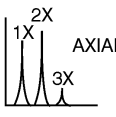
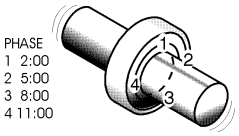
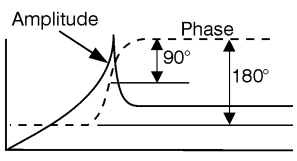
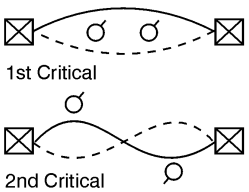
Finally, a remarks column is included in Table 6.0 to provide further explanatory information on machine problem symptoms and diagnostics. For example, there is a warning under the remarks column for the "bent shaft" problem source to be sure and account for transducer orientation when taking axial phase measurements.

It is hoped that this illustrated chart will help users in diagnosing a wide variety of machine problems. Further information is now being researched and field tested which may soon be added to the diagnostic chart as we constantly learn more and more about how machines react when subjected to a whole series of problems and how we can "read" these reactive responses via diagnostic techniques.

Following on the next pages will be separate discussions on each of the problems outlined in Table 6.0. Later, real-world case histories will be presented giving real-world examples of each of these problems.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

TABLE I
ILLUSTRATED VIBRATION DIAGNOSTIC CHART

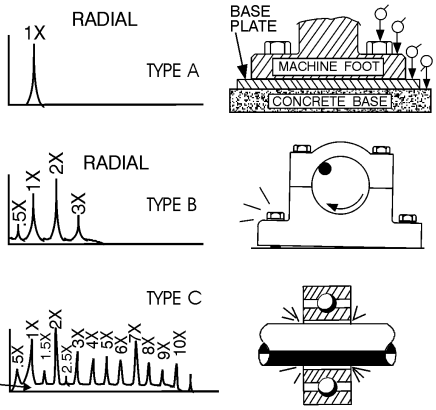
PROBLEM SOURCE	TYPICAL SPECTRUM	PHASE RELATIONSHIP	REMARKS
MASS UNBALANCE **A. FORCE UNBALANCE**			Force Unbalance will be in-phase and steady. Amplitude due to unbalance will increase by the square of speed below first rotor critical (a 3X speed increase = 9X higher vibration). 1X RPM always present and normally dominates spectrum. Can be corrected by placement of only one balance correction weight in one plane at Rotor center of gravity (CG). Approx. 0° phase difference should exist between OB & IB horizontals, as well as between OB & IB verticals. Also, approx. 90° phase difference between horizontal & vertical readings usually occurs on each bearing of unbalanced rotor (±30°).
B. COUPLE UNBALANCE			Couple Unbalance results in 180° out-of-phase motion on same shaft. 1X RPM always present and normally dominates spectrum. Amplitude varies with square of increasing speed below first rotor critical speed. May cause high axial vibration as well as radial. Correction requires placement of balance weights in at least 2 planes. Note that approx. 180° phase difference should exist between OB & IB horizontals, as well as between OB & IB verticals. Also, approx. a 90° difference between the horizontal & vertical phase readings on each bearing usually occurs (±30°)
C. DYNAMIC UNBALANCE			Dynamic Unbalance is the dominant type of unbalance found and is a combination of both force and couple unbalance. 1X RPM dominates the spectrum, and truly requires 2 plane correction. Here, the radial phase difference between outboard and inboard bearings can range anywhere from 0° to 180°. However, the horizontal phase difference should closely match the vertical phase difference, when comparing outboard and inboard bearing measurements (±30°). Secondarily, if unbalance predominates, roughly a 90° phase difference usually results between the horizontal and vertical readings on each bearing (±40°).
D. OVERHUNG ROTOR UNBALANCE			Overhung Rotor Unbalance causes high 1X RPM in both Axial and Radial directions. Axial readings tend to be in-phase whereas radial phase readings might be unsteady. However, the horizontal phase differences will usually match the vertical phase differences on the unbalanced rotor (±30°). Overhung rotors have both force and couple unbalance, each of which will likely require correction. Thus, correction weights will most always have to be placed in 2 planes to counteract both force and couple unbalance.
ECCENTRIC ROTOR			Eccentricity occurs when center of rotation is offset from geometric centerline of a pulley, gear, bearing, motor armature, etc. Largest vibration occurs at 1X RPM of eccentric component in a direction thru centerlines of the two rotors. Comparative horizontal and vertical phase readings usually differ either by 0° or by 180° (each of which indicate straight-line motion). Attempts to balance eccentric rotors often result in reducing vibration in one radial direction, but increasing it in the other radial direction (depending on amount of eccentricity).
BENT SHAFT			Bent shaft problems cause high axial vibration with axial phase differences tending towards 180° on the same machine component. Dominant vibration normally occurs at 1X if bent near shaft center, but at 2X if bent near the coupling. (Be careful to account for transducer orientation for each axial measurement if you reverse probe direction.) Use dial indicators to confirm bent shaft.
MISALIGNMENT **A. ANGULAR MISALIGNMENT**			Angular Misalignment is characterized by high axial vibration, 180° out-of-phase across the coupling. Typically will have high axial vibration with both 1X and 2X RPM. However, not unusual for either 1X, 2X or 3X to dominate. These symptoms may also indicate coupling problems as well. Severe angular misalignment may excite many 1X RPM harmonics. Unlike Mechanical Looseness Type 3, these multiple harmonics do not typically have a raised noise floor on the spectra.
B. PARALLEL MISALIGNMENT			Offset Misalignment has similar vibration symptoms to Angular, but shows high radial vibration which approaches 180° out-of-phase across coupling. 2X often larger than 1X, but its height relative to 1X is often dictated by coupling type and construction. When either Angular or Radial Misalignment becomes severe, they can generate either high amplitude peaks at much higher harmonics (4X-8X), or even a whole series of high frequency harmonics similar in appearance to mechanical looseness. Coupling type and material will often greatly influence the entire spectrum when misalignment is severe. Does not typically have raised noise floor.
C. MISALIGNED BEARING COCKED ON SHAFT			Cocked Bearing will generate considerable axial vibration. Will cause Twisting Motion with approximately 180° phase shift top to bottom and/or side to side as measured in axial direction on same bearing housing. Attempts to align coupling or balance the rotor will not alleviate problem. Bearing usually must be removed and correctly installed.
RESONANCE			Resonance occurs when a Forcing Frequency coincides with a System Natural Frequency, and can cause dramatic amplitude amplification, which might result in premature, or even catastrophic failure. This may be a natural frequency of the rotor, but can often originate from support frame, foundation, gearbox or even drive belts. If a rotor is at or near resonance, it can be almost impossible to balance due to the great phase shift it experiences (90° at resonance; nearly 180° when passes thru). Often requires changing natural frequency to a higher or lower frequency. Natural Frequencies do not generally change with a change in speed which helps facilitate their identification (unless on a large plain bearing machine or on a rotor which has significant overhang).

© COPYRIGHT 1996 - TECHNICAL ASSOCIATES OF CHARLOTTE, P.C. R-0894-4

TABLE I
ILLUSTRATED VIBRATION DIAGNOSTIC CHART

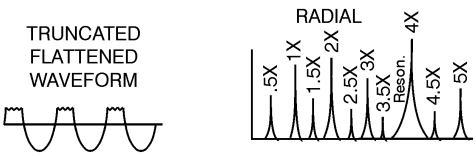
PROBLEM SOURCE	TYPICAL SPECTRUM	PHASE RELATIONSHIP	REMARKS

MECHANICAL LOOSENESS



Mechanical Looseness is indicated by either Type A, B or C vibration spectra.
Type A is caused by Structural looseness/weakness of machine feet, baseplate or foundation; also by deteriorated grouting, loose hold-down bolts at the base; and distortion of the frame or base (i.e., soft foot). Phase analysis may reveal approx. 90° to 180° phase difference between vertical measurements on bolt, machine foot, baseplate or base itself.
Type B is generally caused by loose pillowblock bolts, cracks in frame structure or in bearing pedestal.
Type C is normally generated by improper fit between component parts which will cause many harmonics due to nonlinear response of loose parts to dynamic forces from rotor. Causes a truncation of time waveform and a raised noise floor in the spectrum. Type C is often caused by a bearing liner loose in its cap, a bearing loose and turning on its shaft, excessive clearance in either a sleeve or rolling element bearing, a loose impeller on a shaft, etc. Type C Phase is often unstable and may vary widely from one measurement to next, particularly if rotor shifts position on shaft from one startup to next. Mechanical Looseness is often highly directional and may cause very different readings when comparing levels at 30° increments in radial direction all the way around one bearing housing. Also, note that looseness will often cause subharmonic multiples at exactly 1/2 or 1/3X RPM (.5X, 1.5X, 2.5X, etc.).

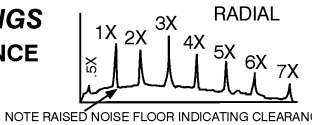
ROTOR RUB



Rotor Rub produces similar spectra to Mechanical Looseness when rotating parts contact stationary components. Rub may be either partial or throughout the entire shaft revolution. Usually generates a series of frequencies, often exciting one or more resonances. Often excites integer fraction subharmonics of running speed (1/2, 1/3, 1/4, 1/5,...1/n), depending on location of rotor natural frequencies. Rotor rub can excite many high frequencies (similar to wide-band noise when chalk is drug along a blackboard). It can be very serious and of short duration if caused by shaft contacting bearing babbitt. A full annular rub throughout an entire shaft revolution can induce "reverse precession" with the rotor whirling at critical speed in a direction opposite shaft rotation (inherently unstable which can lead to catastrophic failure).

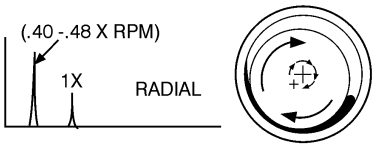
JOURNAL BEARINGS

A. WEAR/CLEARANCE PROBLEMS



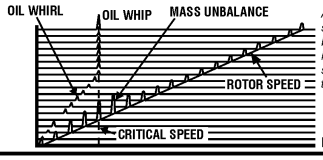
Latter stages of journal bearing wear are normally evidenced by presence of whole series of running speed harmonics (up to 10 or 20). Wiped journal bearings often will allow high vertical amplitudes compared to horizontal, but, may show only one pronounced peak at 1X RPM. Journal bearings with excessive clearance may allow a minor unbalance and/or misalignment to cause high vibration which would be much lower if bearing clearances were set to spec.

B. OIL WHIRL INSTABILITY



Oil Whirl instability occurs at .40 - .48X RPM and is often quite severe. Considered excessive when amplitude exceeds 40% of bearing clearances. Oil Whirl is an oil film excited vibration where deviations in normal operating conditions (attitude angle and eccentricity ratio) cause oil wedge to "push" shaft around within bearing. Destabilizing force in direction of rotation results in a whirl (or forwards precession). Oil Whirl is unstable since it increases centrifugal forces which increase whirl forces. Can cause oil to no longer support shaft and can become unstable when whirl frequency coincides with a rotor natural frequency. Changes in oil viscosity, lube pressure and external preloads can affect oil whirl.

C. OIL WHIP INSTABILITY

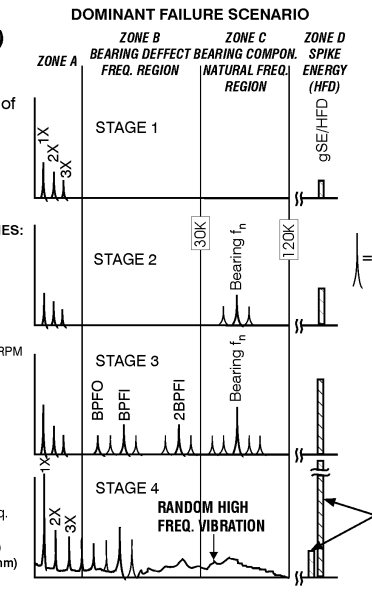


A spectral map showing Oil Whirl becoming Oil Whip instability as shaft speed passes thru 2X critical.

Oil Whip may occur if machine operated at or above 2X rotor critical frequency. When rotor brought up to twice critical speed, whirl will be very close to rotor critical and may cause excessive vibration that oil film may no longer be capable of supporting. Whirl speed will actually "lock onto" rotor critical and this peak will not pass through it even if machine is brought to higher and higher speeds. Produces a lateral forward precessional subharmonic vibration at rotor critical frequency. Inherently unstable which can lead to catastrophic failure.

ROLLING ELEMENT BEARINGS

(4 Failure Stages)



f_n = Natural Frequencies of Installed Bearing Components and Support Structure

BEARING DEFECT FREQUENCIES:

$$BPFI = \frac{N_b}{2}\left(1+\frac{B_d}{P_d}\cos\theta\right) \times RPM$$

$$BPFO = \frac{N_b}{2}\left(1-\frac{B_d}{P_d}\cos\theta\right) \times RPM$$

$$BSF = \frac{P_d}{2B_d}\left[1-\left(\frac{B_d}{P_d}\right)^2(\cos\theta)^2\right] \times RPM$$

$$FTF = \frac{1}{2}\left(1-\frac{B_d}{P_d}\cos\theta\right) \times RPM$$

Where:
BPFI = Inner Race Frequency
BPFO = Outer Race Frequency
BSF = Ball Spin Frequency
FTF = Fund. Train (Cage) Freq.
N_b = Number of Balls or Rollers
B_d = Ball/Roller Diameter (in or mm)
P_d = Bearing Pitch Diameter (in or mm)
θ = Contact Angle (degrees)

4 ROLLING ELEMENT BEARING FAILURE STAGES

STAGE 1: Earliest indications of bearing problems appear in ultrasonic frequencies ranging from about 250,000 - 350,000 Hz; later, as wear increases, usually drops to approximately 20,000 - 60,000 Hz (1,200,000 - 3,600,000 CPM). These are frequencies evaluated by Spike Energy (gSE), HFD(g) and Shock Pulse (dB). For example, spike energy may first appear at about .25 gSE in Stage 1 (actual value depending on measurement location and machine speed). Acquiring high frequency enveloped spectra confirms whether or not bearing is in Failure Stage 1.

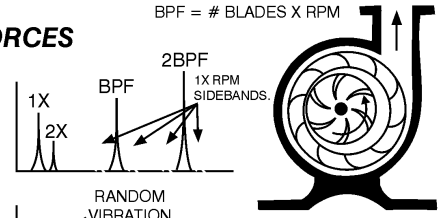
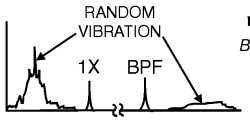
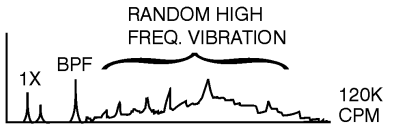
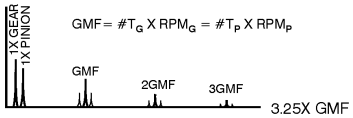
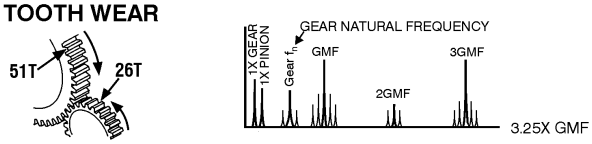
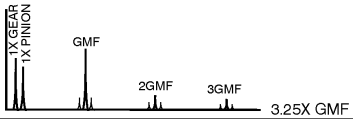
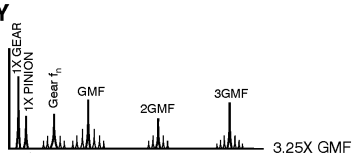
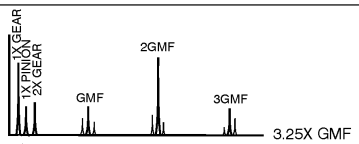
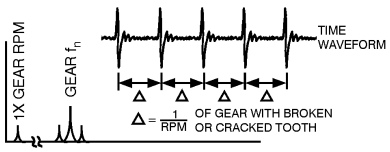
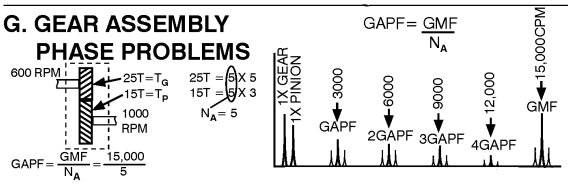
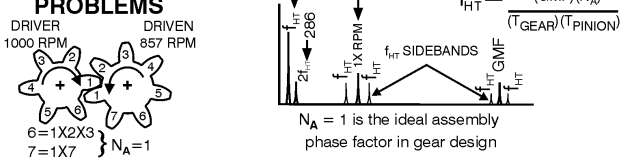
STAGE 2: Slight bearing defects begin to "ring" bearing component natural frequencies (f_n) which predominantly occur in 30K - 120K CPM range. Such natural frequencies may also be resonances of bearing support structures. Sideband frequencies appear above and below natural frequency peak at end of Stage 2. Overall spike energy grows (for example, from .25 to .50 gSE).

STAGE 3: Bearing defect frequencies and harmonics appear. When wear progresses, more defect frequency harmonics appear and number of sidebands grow, both around these and bearing component natural frequencies. Overall spike energy continues to increase (for example, from .5 to over 1 gSE). Wear is now usually visible and may extend throughout periphery of bearing, particularly when many well formed sidebands accompany bearing defect frequency harmonics. High frequency demodulated and enveloped spectra help confirm Stage III. ***Replace bearings now! (independent of bearing defect frequency amplitudes in vibration spectra).***

STAGE 4: Towards the end, amplitude of 1X RPM is even effected. It grows, and normally causes growth of many running speed harmonics. Discrete bearing defect and component natural frequencies actually begin to "disappear" and are replaced by random, broadband high frequency "noise floor". In addition, amplitudes of both high frequency noise floor and spike energy may in fact decrease; but just prior to failure, spike energy and HFD will usually grow to excessive amplitudes.

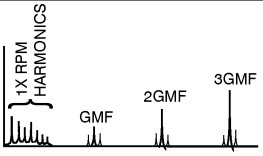
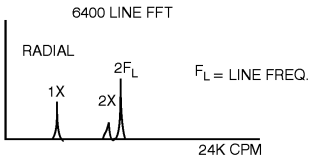
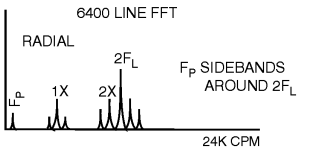
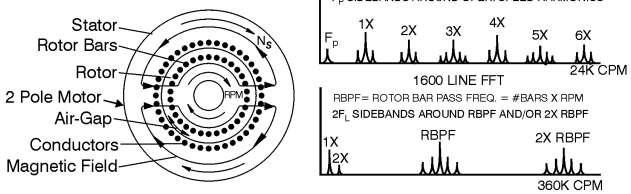
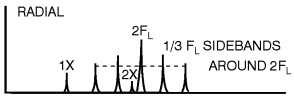
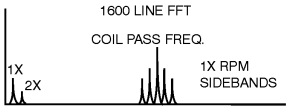
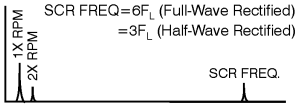
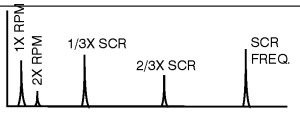
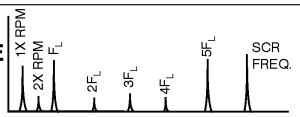
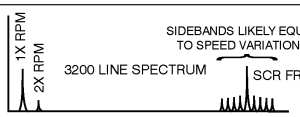
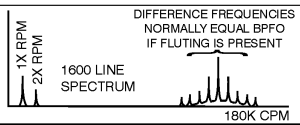
© COPYRIGHT 1996 - TECHNICAL ASSOCIATES OF CHARLOTTE, P.C. R-0894-4

TABLE I
ILLUSTRATED VIBRATION DIAGNOSTIC CHART

PROBLEM SOURCE	TYPICAL SPECTRUM	REMARKS
HYDRAULIC AND AERODYNAMIC FORCES		
A. BLADE PASS & VANE PASS		Blade Pass Frequency (BPF) = No. of Blades (or Vanes) X RPM. This frequency is inherent in pumps, fans and compressors, and normally does not present a problem. However, large amplitude BPF (and harmonics) can be generated in pump if gap between rotating vanes and stationary diffusers is not equal all the way around. Also, BPF (or harmonic) sometimes can coincide with a system natural frequency causing high vibration. High BPF can be generated if impeller wear ring seizes on shaft, or if welds fastening diffuser vanes fail. Also, high BPF can be caused by abrupt bends in pipe (or duct), obstructions which disturb flow, damper settings or if pump or fan rotor is positioned eccentrically within housing.
B. FLOW TURBULENCE		Flow Turbulence often occurs in blowers due to variations in pressure or velocity of the air passing thru the fan or connected ductwork. This flow disruption causes turbulence which will generate random, low frequency vibration, typically in the range of 50 to 2000 CPM. If surging occurs within a compressor, random broadband high frequency vibration can occur. Excessive turbulence can also excite broadband high frequency.
C. CAVITATION		Cavitation normally generates random, higher frequency broadband energy which is sometimes superimposed with blade pass frequency harmonics. Normally indicates insufficient suction pressure (starvation). Cavitation can be quite destructive to pump internals if left uncorrected. It can particularly erode impeller vanes. When present, it often sounds as if "gravel" is passing thru pump. Cavitation is usually caused by insufficient inlet flow. Can occur during one survey, and be absent the next survey (if changes in suction valve settings are made).
GEARS		
A. NORMAL SPECTRUM		Normal Spectrum shows Gear & Pinion Speeds, along with Gear Mesh Frequency (GMF) and very small GMF harmonics. GMF harmonics commonly will have running speed sidebands around them. All peaks are of low amplitude, and no natural frequencies of gears are excited. F_{MAX} recomended at 3.25X GMF (minimum) when # teeth are known. If tooth count is not known, set F_{MAX} at 200X RPM on each shaft.
B. TOOTH WEAR		Key indicator of Tooth Wear is excitation of Gear Natural Frequency (f_n), along with sidebands around it spaced at the running speed of the bad gear. Gear Mesh Frequency (GMF) may or may not change in amplitude, although high amplitude sidebands and number of sidebands surrounding GMF usually occur when wear is noticeable. Sidebands may be better wear indicator than GMF frequencies themselves. Also, high amplitudes commonly occur at either 2XGMF or at 3XGMF (esp. 3XGMF), even when GMF amplitude is acceptable.
C. TOOTH LOAD		Gear Mesh Frequencies are often very sensitive to load. High GMF amplitudes do not necessarily indicate a problem, particularly if sideband frequencies remain low level, and no gear natural frequencies are excited. Each Analysis should be performed with system at maximum operating load for meaningful spectral comparisons.
D. GEAR ECCENTRICITY AND BACKLASH		Fairly high amplitude sidebands around GMF harmonics often suggest gear eccentricity, backlash, or non-parallel shafts which allow the rotation of one gear to "modulate" either the GMF amplitude or the running speed of the other gear. The gear with the problem is indicated by the spacing of the sideband frequencies.Also, 1X RPM level of eccentric gear will normally be high if eccentricity is the dominant problem. Improper backlash normally excites GMF harmonics and Gear Natural Frequency, both of which will be sidebanded at 1X RPM. GMF amplitudes will often decrease with increasing load if backlash is the problem.
E. GEAR MISALIGNMENT		Gear Misalignment almost always excites second order or higher GMF harmonics which are sidebanded at running speed. Often will show only small amplitude 1X GMF, but much higher levels at 2X or 3X GMF. Important to set F_{MAX} high enough to capture at least 3 GMF harmonics. Also, sidebands around 2XGMF will often be spaced at 2X RPM. Note that sideband amplitudes often are not equal on left and right side of GMF and GMF harmonics due to the tooth misalignment. Causes uneven wear pattern.
F. CRACKED/BROKEN TOOTH		A Cracked or Broken Tooth will generate a high amplitude at 1X RPM of this gear only in the time waveform, plus it will excite gear natural frequency (f_n) sidebanded at its running speed. It is best detected in Time Waveform which will show a pronounced spike every time the problem tooth tries to mesh with teeth on the mating gear. Time between impacts (Δ) will correspond to 1/RPM of gear with the problem. Amplitudes of Impact Spikes in Time Waveform often will be 10X to 20X higher than that at 1X RPM in the FFT!
G. GEAR ASSEMBLY PHASE PROBLEMS	 GAPF= 3000 CPM = 0.20X GMF **(FRACTIONAL GMF)**	Gear Assembly Phase Freq. (GAPF) can result in Fractional Gear Mesh Frequencies (if N_A>1). It literally means (T_G/N_A) gear teeth will contact (T_P/N_A) pinion teeth and will generate N_A wear patterns, where N_A in a given tooth combination equals the product of prime factors common to the number of teeth on the gear and pinion (N_A = Assembly Phase Factor). GAPF (or harmonics) can show up right from the beginning if there were manufacturing problems. Also, its sudden appearance in a periodic survey spectrum can indicate damage if contaminate particles pass through the mesh, resulting in damage to the teeth in mesh at the time of ingestion just as they enter and leave meshing or that gears have been reoriented.
H. HUNTING TOOTH PROBLEMS	 f_{HT}= $\frac{(6X\ 1000)(1)}{(6)(7)}$ = $\frac{1000}{7}$ = 143 CPM (One Pulse Per 7 Pinion Revolutions)	Hunting Tooth Frequency (f_{HT}) occurs when faults are present on both the gear and pinion which might have occurred during the manufacturing process, due to mishandling, or in the field. It can cause quite high vibration, but since it occurs at low frequencies predominately less than 600 CPM, it is often missed. A gear set with this tooth repeat problem normally emits a "growling" sound from the drive. The maximum effect occurs when the faulty pinion and gear teeth both enter mesh at the same time (on some drives, this may occur only 1 of every 10 to 20 revolutions, depending on the f_{HT} formula). Note that T_{GEAR} and T_{PINION} refer to number of teeth on the gear and pinion, respectively. N_A is the Assembly Phase Factor defined above. Will often modulate both GMF and Gear RPM peaks.

© COPYRIGHT 1996 - TECHNICAL ASSOCIATES OF CHARLOTTE, P.C. R-0894-4

TABLE I
ILLUSTRATED VIBRATION DIAGNOSTIC CHART

PROBLEM SOURCE	TYPICAL SPECTRUM	REMARKS
GEARS (CONTINUED)		
I. LOOSE BEARING FIT		Excessive Clearance of bearings supporting the gears can not only excite many running speed harmonics, but will often cause high amplitude response at GMF, 2GMF and/or 3GMF. These high GMF amplitudes are actually a response to, and not the cause of, looseness within the bearings supporting the gearing. Such excessive clearance can be caused either by extensive bearing wear or by improper bearing fit onto the journal during installation. Left uncorrected, it can cause excessive gear wear and damage to other components.
AC INDUCTION MOTORS		
A. STATOR ECCENTRICITY, SHORTED LAMINATIONS OR LOOSE IRON		Stator problems generate high vibration at 2X line frequency ($2F_L$). Stator eccentricity produces uneven stationary air gap between rotor and stator which produces very directional vibration. Differential Air Gap should not exceed 5% for induction motors and 10% for synchronous motors. Soft foot and warped bases can produce an eccentric stator. Loose iron is due to stator support weakness or looseness. Shorted stator laminations can cause uneven, localized heating which can distort the stator itself. This produces thermally-induced vibration which can significantly grow with operating time causing stator distortion and static air gap problems.
B. ECCENTRIC ROTOR (Variable Air Gap) F_L = Electrical Line Freq. N_S = Synch. Speed = $\frac{120F_L}{P}$ F_S = Slip Freq. = N_S – RPM F_P = Pole Pass Freq. = F_S X P P = #Poles		Eccentric Rotors produce a rotating variable air gap between the rotor and stator which induces pulsating vibration (normally between $2F_L$ and closest running speed harmonic). Often requires "zoom" spectrum to separate $2F_L$ and running speed harmonic. Eccentric rotors generate $2F_L$ surrounded by Pole Pass frequency sidebands (F_P), as well as F_P sidebands around running speed. F_P appears itself at low frequency (Pole Pass Frequency = Slip Frequency X #Poles). Common values of F_P range from about 20 to 120 CPM (0.3 - 2.0 Hz). Soft foot or misalignment often induces a variable air gap due to distortion (actually a mechanical problem; not electrical).
C. ROTOR PROBLEMS		Broken or Cracked rotor bars or shorting rings; bad joints between rotor bars and shorting rings; or shorted rotor laminations will produce high 1X running speed vibration with pole pass frequency sidebands (F_P). In addition, these problems will often generate F_P sidebands around the second, third, fourth and fifth running speed harmonics. Loose or open rotor bars are indicated by 2X line freq. ($2F_L$) sidebands surrounding Rotor Bar Pass Frequency (RBPF) and/or its harmonics (RBPF = Number of Bars X RPM). Often will cause high levels at 2X RBPF, with only a small amplitude at 1X RBPF. Electrically induced arcing between loose rotor bars and end rings will often show high levels at 2X RBPF (with $2F_L$ sidebands); but little or no increase in amplitudes at 1X RBPF.
D. PHASING PROBLEM (Loose Connector)		Phasing problems due to loose or broken connectors can cause excessive vibration at 2X Line Freq. ($2F_L$) which will have sidebands around it spaced at 1/3 Line Freq. ($1/3\ F_L$). Levels at $2F_L$ can exceed 1.0 in/sec if left uncorrected. This is particularly a problem if the defective connector is only sporadically making contact. Loose or broken connectors must be repaired to prevent catastrophic failure.
AC SYNCHRONOUS MOTORS **(Loose Stator Coils)**		Loose stator coils in synchronous motors will generate fairly high vibration at Coil Pass Freq. (CPF) which equals the number of stator coils X RPM (#Stator Coils = #Poles X #Coils/Pole). The Coil Pass Frequency will be surrounded by 1X RPM sidebands. Synchronous motor problems may also be indicated by high amplitude peaks at approx. 60,000 to 90,000 CPM, accompanied by $2F_L$ sidebands. Take at least one spectrum up to 90,000 CPM on each motor bearing housing.
DC MOTORS AND CONTROLS **A. NORMAL SPECTRUM**		Many DC Motor and Control Problems can be detected by vibration analysis. Full-wave rectified, motors (6 SCR's) generate a signal at 6X Line Frequency ($6F_L$=360 Hz=21,600 CPM); while half-wave rectified DC motors (3 SCR's) generate 3X Line Freq. ($3F_L$=180 Hz=10,800 CPM). The SCR firing Frequency is normally present in a DC Motor Spectrum, but at low amplitude. Note the absence of other peaks at multiples of F_L.
B. BROKEN ARMATURE WINDINGS, GROUNDING PROBLEMS OR FAULTY SYSTEM TUNING		When DC Motor spectra are dominated by high levels at SCR or 2X SCR, this normally indicates either Broken Motor Windings or Faulty Tuning of the Electrical Control System. Proper tuning alone can lower vibration at SCR and 2X SCR significantly if control problems predominate. High amplitudes at these frequencies would normally be above approximately .10 in/sec, peak at 1 X SCR and about .04 in/sec at 2 X SCR Firing Freq.
C. FAULTY FIRING CARD OR BLOWN FUSE		When one firing card fails to fire, then 1/3 of power is lost, and can cause repeated momentary speed changes in the motor. This can lead to high amplitudes at 1/3X and 2/3X SCR Frequency (1/3X SCR Freq. = 1X F_L for half-wave rectified, but 2X F_L for a full-wave rectified SCR). Caution: Card/SCR configuration should be known before troubleshooting motor (#SCR's, #Firing Cards, etc.).
D. FAULTY SCR, SHORTED CONTROL CARD, LOOSE CONNECTIONS AND/OR BLOWN FUSE		Faulty SCR's, Shorted Control Cards and/or Loose Connections can generate noticeable amplitude peaks at many combinations of line frequency (F_L) and SCR firing frequency. Normally, 1 bad SCR can cause high levels at F_L and/or $5F_L$ in 6 SCR motors. The point to be made is that neither F_L, $2F_L$, $4F_L$ nor $5F_L$ should be present in DC Motor spectra.
E. FAULTY COMPARITOR CARD		Faulty Comparitor Cards cause problems with RPM fluctuation or "hunting". This causes a constant collapsing and regenerating of the magnetic field. These sidebands often approximate the RPM fluctuation and require a high resolution FFT to even detect them. Such sidebands could also be due to generation and regeneration of the magnetic field.
F. ELECTRICAL CURRENT PASSAGE THRU DC MOTOR BEARINGS		Electrically-induced Fluting is normally detected by a series of difference frequencies with the spacing most often at the outer race defect frequency (BPFO), even if such fluting is present on both the outer and inner races. They most often show up in a range centered at about 100,000 to 150,000 CPM. A 180K CPM spectrum with 1600 lines is recommended for detection with measurements on both the OB and IB DC motor bearings.

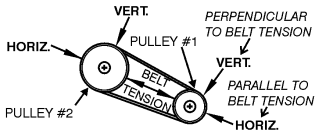
© COPYRIGHT 1996 - TECHNICAL ASSOCIATES OF CHARLOTTE, P.C. R-0894-4

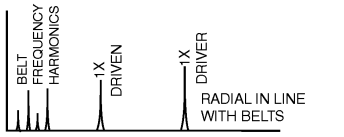
TABLE I
ILLUSTRATED VIBRATION DIAGNOSTIC CHART

PROBLEM SOURCE	TYPICAL SPECTRUM	REMARKS

BELT DRIVE PROBLEMS

A. WORN, LOOSE OR MISMATCHED BELTS





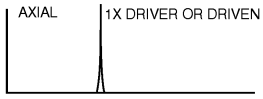
PITCH DIAM$_1$ X RPM$_1$ = PITCH DIAM$_2$ X RPM$_2$

BELT FREQ. = $\frac{3.142 \text{ X PULLEY RPM X PITCH DIAM.}}{\text{BELT LENGTH}}$

TIMING BELT FREQ. = BELT FREQ. X #BELT TEETH
= PULLEY RPM X #PULLEY TEETH

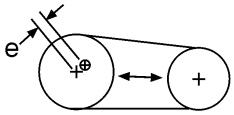
Belt frequencies are below the RPM of either the motor or the driven machine. When they are worn, loose or mismatched, they normally cause 3 to 4 multiples of belt frequency. Often 2X belt freq. is the dominant peak. Amplitudes are normally unsteady, sometimes pulsing with either driver or driven RPM. On timing belt drives, wear or pulley misalignment is indicated by high amplitudes at the Timing Belt Frequency. Chain drives will indicate problems at Chain Pass Frequency which equals #Sprocket Teeth X RPM.

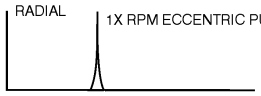
B. BELT/PULLEY MISALIGNMENT



Misalignment of pulley produces high vibration at 1X RPM predominantly in the axial direction. The ratio of amplitudes of driver to driven RPM depends on where the data is taken, as well as on relative mass and frame stiffness. Often with pulley misalignment, the highest axial vibration on the motor will be at fan RPM, or vice versa. Can be confirmed by phase measurements by setting Phase Filter at RPM of pulley with highest axial amplitude; then compare phase at this particular frequency on each rotor in the axial direction.

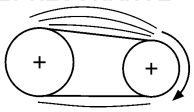
C. ECCENTRIC PULLEYS

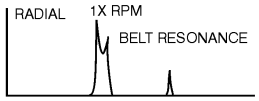




Eccentric pulleys cause high vibration at 1X RPM of the eccentric pulley. The amplitude is normally highest in line with the belts, and should show up on both driver and driven bearings. It is sometimes possible to balance eccentric pulleys by attaching washers to taper-lock bolts. However, even if balanced, the eccentricity will still induce vibration and reversible fatigue stresses in the belt. Pulley eccentricity can be confirmed by phase analysis showing horizontal & vertical phase differences of nearly 0° or 180°.

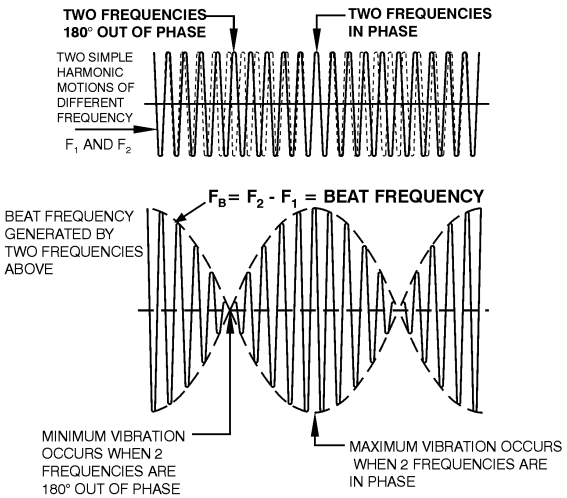
D. BELT RESONANCE





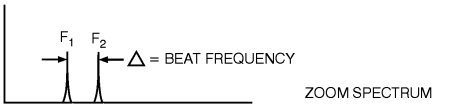
Belt Resonance can cause high amplitudes if the belt natural frequency should happen to approach, or coincide with, either the motor or driven RPM. Belt natural frequency can be altered by changing either the belt tension, belt length or cross section. Can be detected by tensioning and then releasing belt while measuring the response on pulleys or bearings. However, when operating, belt natural frequencies will tend to be slightly higher on the tight side and lower on the slack side.

BEAT VIBRATION

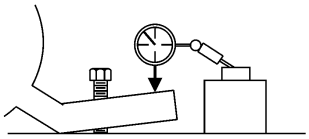


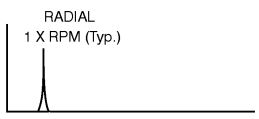
A Beat Frequency is the result of two closely spaced frequencies going into and out of synchronization with one another. The wideband spectrum normally will show one peak pulsating up and down. When you zoom into this peak (lower spectrum below), it actually shows two closely spaced peaks. The difference in these two peaks (F_2-F_1) is the beat frequency which appears itself in the wideband spectrum. The beat frequency is not commonly seen in normal frequency range measurements since it is inherently low frequency, usually ranging from only approximately 5 to 100 CPM.

Maximum vibration will result when the time waveform of one frequency (F_1) comes into phase with the waveform of the other frequency (F_2). Minimum vibration occurs when waveforms of these two frequencies line up 180° out of phase.



SOFT FOOT, SPRUNG FOOT AND FOOT-RELATED RESONANCE





"***Soft Foot***" occurs when a machine's foot or frame deflects greatly when a hold-down bolt is loosened to hand tightness, causing the foot to rise more than approximately .002 - .003 inch. This does not always cause a great vibration increase. However, it can do so if the soft foot affects alignment or motor air gap concentricity.

"***Sprung Foot***" can cause great frame distortion, resulting in increased vibration, force and stress in the frame, bearing housing, etc. This can occur when a hold-down bolt is forceably torqued down on the sprung foot in an attempt to level the foot.

"***Foot-Related Resonance***" can cause dramatic amplitude increases from 5X to 15X or more, as compared with that when the bolt (or combination of bolts) is loosened to hand tightness. When tight, this bolt can notably change the natural frequency of the foot or machine frame itself.

Soft Foot, Sprung Foot or Foot-Related Resonance most often affects vibration at 1X RPM, but can also do so at 2X RPM, 3X RPM, 2X line frequency, blade pass frequency, etc. (particularly Foot-Related Resonance).

© COPYRIGHT 1996 - TECHNICAL ASSOCIATES OF CHARLOTTE, P.C. R-0894-4

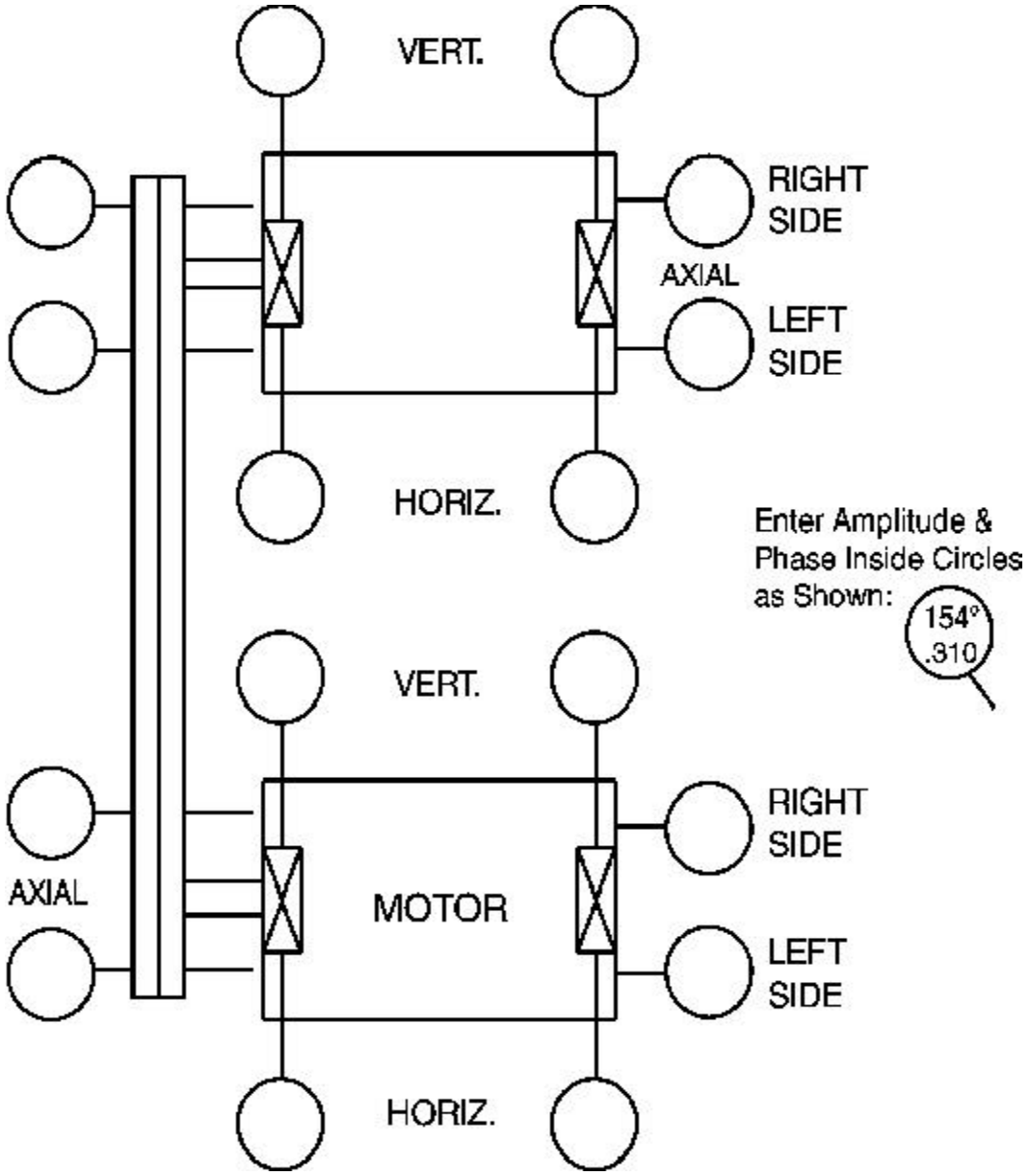


FIGURE 6.0A
PHASE ANALYSIS DIAGRAM BDB-1

© Copyright 2001 Technical Associates Of Charlotte, P.C.

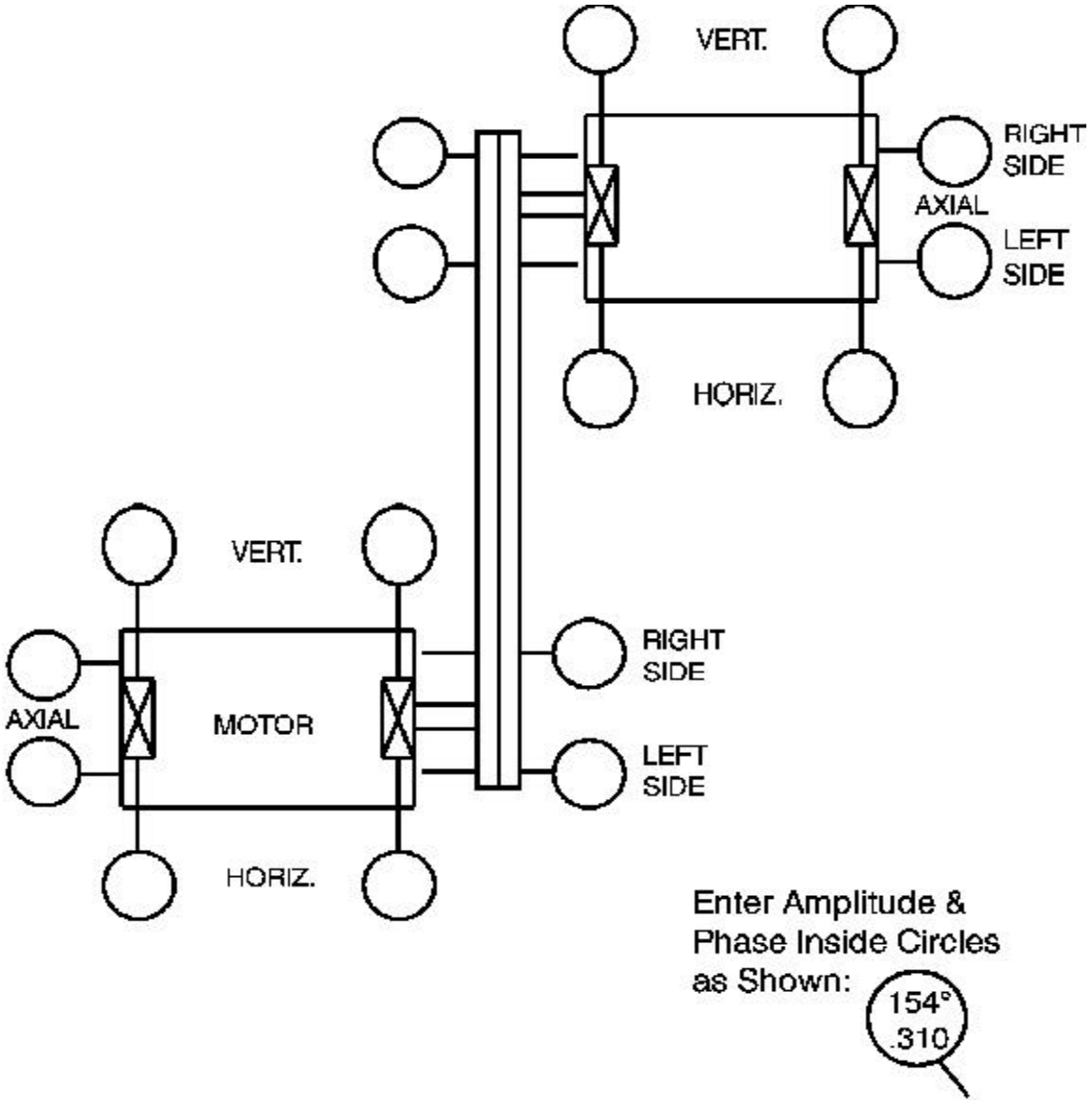


FIGURE 6.0B
PHASE ANALYSIS DIAGRAM BDB-2

© Copyright 2001 Technical Associates Of Charlotte, P.C.

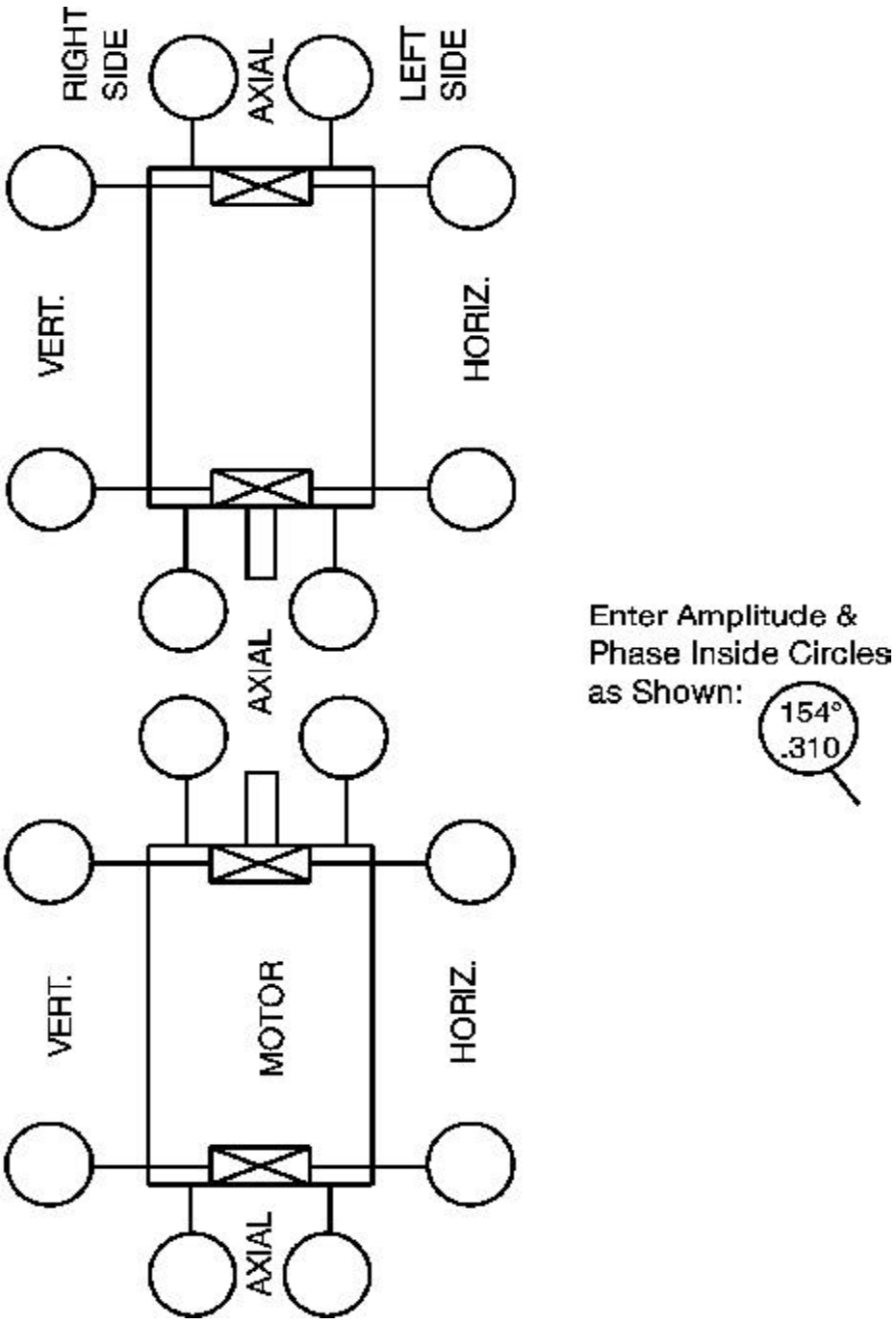


FIGURE 6.0C
PHASE ANALYSIS DIAGRAM DC-1

© Copyright 2001 Technical Associates Of Charlotte, P.C.

6.01 MASS UNBALANCE

Unbalance occurs when the mass centerline does not coincide with the shaft centerline as shown in Figures 6.01A thru 6.01D. Some degree of unbalance exists in all rotors whether they are a cooling tower fan or a precision grinding wheel. The key is to know how much unbalance is acceptable for the particular type of machine at its specific operating speed which will be discussed in Section 6.015 “Allowable Residual Unbalance”.

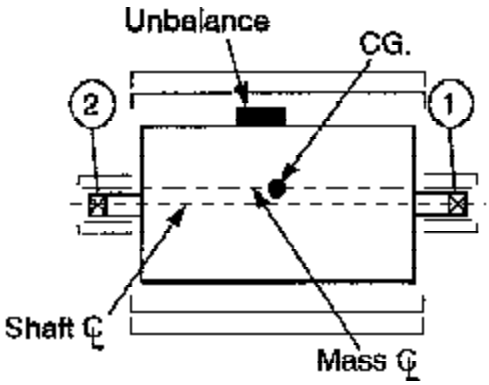


FIGURE 6.01A
FORCE UNBALANCE

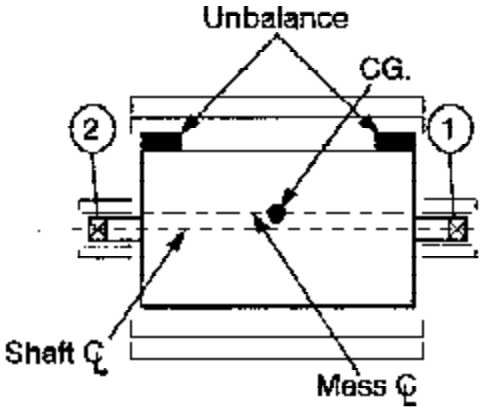


FIGURE 6.01B
FORCE UNBALANCE ALSO

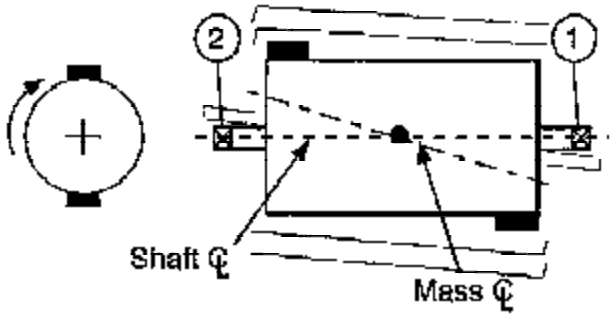


FIGURE 6.01C
COUPLE UNBALANCE

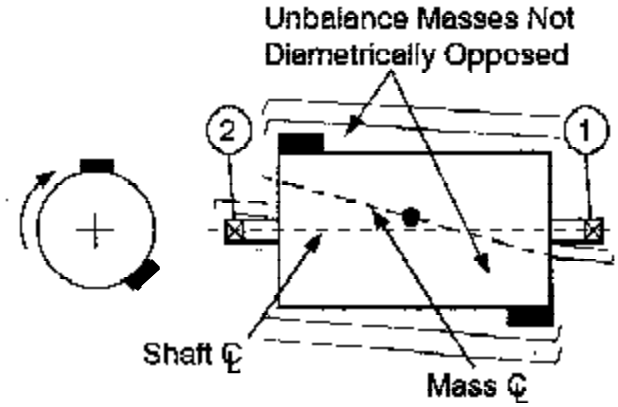


FIGURE 6.01D
DYNAMIC UNBALANCE

© Copyright 2001 Technical Associates Of Charlotte, P.C.

Unbalanced rotors exhibit each of the following characteristics:

1. Unbalance is always indicated by high vibration at 1X RPM of the unbalanced part (but, vibration at 1X RPM is not always unbalance). Normally, this 1X RPM peak will dominate the spectrum.

2. The amplitude at 1X RPM will normally be greater than or equal to 80% of the overall amplitude when the problem is limited to unbalance (may be only 50% to 80% if other problems exist in addition to unbalance).

3. The amplitude of vibration is proportional to how far the mass center is displaced from the shaft center. For example, when operating below the first rotor critical speed, amplitude will vary with the square of RPM (that is, tripling the speed will result in an increase in unbalance vibration by a factor of 9 times).

4. Mass unbalance generates a uniform rotating force which is continually changing direction, but is evenly applied in all radial directions. As a result, the shaft and supporting bearings tend to move in somewhat a circular orbit. However, due to the fact that vertical bearing stiffness is normally higher than that in the horizontal direction, the normal response is a slightly elliptical orbit. Subsequently, horizontal vibration is normally somewhat higher than that in the vertical commonly ranging between 2 and 3 times higher. When the ratio of horizonal to vertical is higher than about 6 to 1, it normally indicates other problems, particularly resonance.

5. When unbalance dominates over other problems, there will normally be about a 90° phase difference between horizontal and vertical directions on a bearing (±30°). Therefore, if there is a high vibration at 1X RPM, but this phase difference is either 0° or close to 180°, it normally points to another problem source such as eccentricity.

6. Probably an even greater indicator of unbalance than the approximately 90° phase shift between horizontal and vertical is the fact that when significant unbalance exists, the horizontal phase difference between outboard and inboard bearings should be close to the difference in phase in the vertical direction. That is, instead of comparing horizontal and vertical phase on the same bearing, compare outboard and inboard horizontal phase difference with outboard and inboard vertical phase difference. For example, please refer to Table A of Figure 6.01E which shows a machine having dominant force unbalance. Note that the horizontal phase difference between the #1 and #2 bearings is about 5° (30° minus 25°) compared to a vertical phase difference of about 10° (120° minus 110°). Similarly, over on the pump, the horizontal phase difference (position 3) is about 10° and the vertical phase difference is about 15°. This is the expected phase response with dominant force unbalance.

7. When unbalance is dominant, radial vibration (horizontal and vertical) will normally be quite much higher than that in the axial direction (except for overhung rotors which will be discussed in Section 6.014).

8. Unbalanced rotors normally exhibit steady and repeatable phase in radial directions. When the rotor is trim balanced, the phase can begin to “dwell” back and forth under a strobe light as you achieve a better and better balance, particularly if other problems are present. However, if there is high unbalance, and other problems are not significant, the phase should be steady and repeatable.

9. The effects of unbalance may sometimes be amplified by resonance. This will be discussed in Section 6.05.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

10. Unbalance can be a great contributor to high looseness vibration. In fact, on a rotor with unbalance and looseness, if it is possible to balance the rotor, this may substantially reduce the looseness vibration although it will often return when even the least little unbalance component returns. Often, it is not even possible to balance rotors having noticeable looseness.

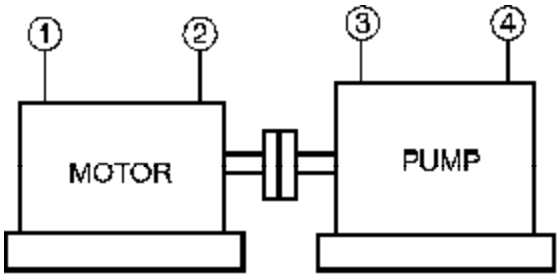


TABLE A.
PHASE AT 1X RPM FOR A MACHINE
WITH DOMINANT FORCE UNBALANCE

DIR.	1	2	3	4
A	60°	*70°	60°	*80°
H	30°	25°	30°	40°
V	120°	110°	120°	135°

* Corrected Phase Values Accounting For 180° Orientation of Accelerometer.

TABLE B.
PHASE AT 1X RPM FOR A MACHINE
WITH DOMINANT COUPLE UNBALANCE

DIR.	1	2	3	4
A	60°	*70°	60°	*80°
H	30°	210°	200°	180°
V	120°	295°	280°	300°

* Corrected Phase Values Accounting For 180° Orientation of Accelerometer.

TABLE C.
PHASE AT 1X RPM FOR A MACHINE
WITH SIGNIFICANT DYNAMIC UNBALANCE

DIR.	1	2	3	4
A	60°	*70°	60°	*80°
H	30°	90°	80°	70°
V	120°	180°	170°	165°

* Corrected Phase Values Accounting For 180° Orientation of Accelerometer.

FIGURE 6.01E
TYPICAL PHASE MEASUREMENTS WHICH WOULD INDICATE EITHER FORCE, COUPLE OR DYNAMIC UNBALANCE

© Copyright 2001 Technical Associates Of Charlotte, P.C.

There are 3 major types of unbalance including force, couple and dynamic unbalance which will be discussed in Sections 6.011, 6.012 and 6.013, respectively, which follow:

6.011 Force Unbalance:

Force unbalance is sometimes known as “static unbalance”. Force unbalance is a condition where the mass centerline is displaced from and parallel to the shaft centerline as shown in Figure 6.01A. This is the type of unbalance that has been classically corrected for many years by placing a fan rotor on knife edges or within its bearings and allowing it to “roll to the bottom”. That is, when the fan wheel is released, if the heavy spot is angularly displaced from the bottom (6:00 position), it would tend to roll to the bottom hopefully ending up in the 6:00 position, assuming the rotor was sufficiently free within its bearings to rotate. So-called correction of this force unbalance was then accomplished by placing a weight opposite this location (or at about 12:00).

Actually, there are two types of force unbalance as shown in Figure 6.01A and Figure 6.01B. In the case of Figure 6.01A, only one heavy spot exists and is located close to the rotor center of gravity (CG). This is corrected by simply placing an equal weight 180° opposite the angular position of the heavy spot. Figure 6.01B likewise illustrates force unbalance even though it shows heavy spots acting on both the outboard and inboard planes (angularly parallel to one another). In this case, it can either be corrected by placing correction weights either at the CG, or by placing equal and opposite weights at each of the two planes (if corrected at the CG, it would of course require double the correction weight in this case).

Characteristics common to force unbalance can be summarized as follows:

1. Approximately the same unbalance forces at 1X RPM are normally present both on the outboard and inboard rotor bearing housings (however, horizontal and vertical responses may differ somewhat depending on the support stiffness in each direction).

2. With pure force unbalance, the outboard horizontal phase will equal the inboard horizontal phase on the same shaft (that is, if the horizontal phase on the outboard bearing were at 6:00, the inboard reading should likewise be about 6:00 since the two shaft ends are moving together).

3. Likewise, the outboard vertical phase should approximately equal the inboard vertical phase on the same shaft.

4. Force unbalance only requires a single plane correction with the counterweight acting through the rotor CG.

5. The difference in horizontal outboard and inboard phase should approximately equal the phase difference in outboard and inboard vertical phase and the phase change across the coupling should be small (less than 60° to 90°) if force unbalance were dominant.

6.012 Couple Unbalance:

Couple unbalance is a condition where the mass centerline axis intersects the shaft centerline axis at the rotor’s center of gravity as shown in Figure 6.01F. Here, a couple is created by equal heavy spots at each end of the rotor, but 180° opposite each other. Significant couple unbalance can introduce severe instability to the rotor causing it to wobble back and forth (like a “seesaw” with the fulcrum at the rotor CG).

© Copyright 2001 Technical Associates Of Charlotte, P.C.

Couple unbalance exhibits each of the following characteristics:

1. In pure couple unbalance, the rotor is statically balanced and will not roll to the bottom when the rotor is placed on knife edges. That is, referring to Figure 6.01C, since the heavy spot at position 1 is equal to that at position 2, this meets the requirement for force or static balance. Still, this rotor will also generate considerable vibration at 1X RPM.

2. Couple unbalance generates high amplitude vibration at 1X RPM on both the outboard and inboard bearing housings, but it may be somewhat higher on one bearing than on the other.

3. Substantial couple unbalance can sometimes generate high axial vibration.

4. The horizontal phase difference between the outboard and inboard bearings will approximate 180° (that is, if the outboard horizontal phase were at 6:00, then the inboard horizontal phase will probably be about 12:00 since the two ends are moving opposite each other in a rocking motion).

5. Similarly, the vertical phase difference between outboard and inboard bearings will approximate 180°.

6. Refer to Table B of Figure 6.01E illustrating how phase should react to couple unbalance. Note the 180° phase difference between position 1 and 2 horizontal (210° - 30°), and the 175° phase difference between position 1 and 2 vertical (295° - 120°). This shows that if the problem is couple unbalance (and not misalignment), both the horizontal and vertical phase differences should roughly be equal to one another - both approximately 180° difference between the outboard and inboard bearings.

6.013 Dynamic Unbalance:

Dynamic unbalance is by far the most common type of unbalance as compared to either purely force or couple unbalance and is defined as “that condition in which the mass centerline is neither parallel to nor intersects the shaft centerline axis”. In essence, dynamic unbalance is a combination of both force and couple unbalance. It requires correction in at least 2 planes perpendicular to the shaft centerline axis.

Dynamic unbalance exhibits each of the following characteristics:

1. Dynamic unbalance generates high vibration at 1X RPM, but the amplitude on the outboard bearing may be somewhat different than that on the inboard bearing housing. Still, they should be within the same order of magnitude, or below about 3 to 1 assuming there are no other significant problems present.

2. Like force and couple unbalance, phase is still steady and repeatable when dynamic unbalance dominates.

3. Although the horizontal phase difference between outboard and inboard bearings could be anything from 0° to 180°, this difference should still approximately equal the vertical phase difference. For example, if the horizontal phase difference was about 60°, the vertical phase difference should likewise be about 60° (±30°) as illustrated in Table C of Figure 6.01E. Here, in this example, notice that the phase difference in both horizontal and vertical directions at positions 1 and 2 is about 60° and that the phase difference across the coupling does not approach 180°. Dynamic unbalance requires correction in at least 2 planes.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

4. Whether or not force or couple balance dominates, the horizontal phase difference at bearings 1 and 2 should approximately equal the vertical phase difference at these two bearings (if the horizontal phase difference is about 150° showing high couple unbalance, the vertical phase difference will approximate 150° as well) **.**

6.014 Overhung Rotor Unbalance:

Figure 6.01F shows an overhung rotor. In this case, the driven rotor is placed outboard of bearings 1 & 2 (rotors which are placed between bearings are known as simply supported rotors). Overhung rotors can cause some interesting vibration symptoms and often can present real problems to the analyst when he attempts to balance one. Overhung rotors display the following characteristics:

1. Overhung rotors can generate large axial forces at 1X RPM which can cause axial vibration to be equal to or greater than radial vibration amplitudes.

2. Overhung rotors often generate a high degree of couple unbalance in addition to force unbalance, both of which must be corrected.

3. Referring to Figure 6.01F, for pure unbalance of an overhung rotor, the axial phase at bearing 1 will approximately equal that at bearing 2 (±30°). Here again, this phase difference depends on how dominant the unbalance problem is as compared to others such as misalignment, resonance, etc.

4. Normally, overhung rotor unbalance can be corrected by first taking care of the force unbalance component which would leave the remainder as couple unbalance with phase differences approaching 180°. The couple component would then require placement of correction weights in 2 planes 180° opposite one another.

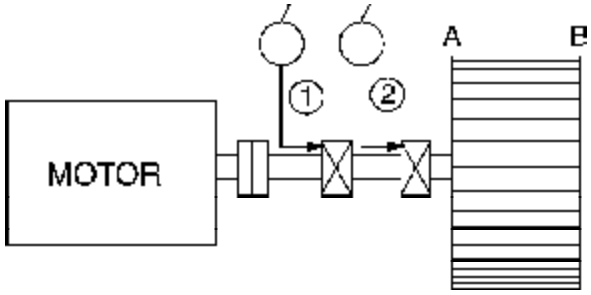


FIGURE 6.01F
BALANCING OF AN OVERHUNG ROTOR

6.0141 Summary of Procedures for Balancing Overhung Rotors

Overhung rotors are machine configurations like that shown in Figure 6.01G where the fan wheel to be balanced is outboard of its two supporting bearings. This configuration is very often found with machines such as blowers, pumps, etc. Because the planes where balance correction weights must be attached are outside the supporting bearings, these rotors will often not respond to standard single and two-plane balancing techniques. In addition, because the unbalance planes are outside the support bearings, even a static unbalance alone will create a couple unbalance proportional to the distance of the unbalance plane from the rotor CG. Therefore, when attempting to balance overhung rotors, the analyst needs to take into account both static and couple unbalance forces, and treat them accordingly.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

When balancing an overhung rotor, one of the two following procedures should be taken:

1. Balancing Overhung Rotors by Classic Single-Plane Static-Couple Method:

Figure 6.01G helps explain methods of balancing overhung rotors. Classically, Bearing A is most sensitive to static unbalance whereas the bearing farthest from the fan wheel to be balanced (Bearing B) is most sensitive to couple unbalance. Since Plane 1 is closest to the rotor center of gravity (CG), static corrections should be made in this plane while measuring the response on Bearing A. On the other hand, measurements should be made on Bearing B when making couple corrections in Plane 2. However, placing a trial weight in Plane 2 will destroy the static balance achieved at Bearing A. Therefore, in order to maintain the static balance at Bearing A, a trial weight placement which will generate a couple must be used. Thus, a trial weight of identical size should be placed in Plane 1 at an angle 180° opposite the trial weight location in Plane 2.

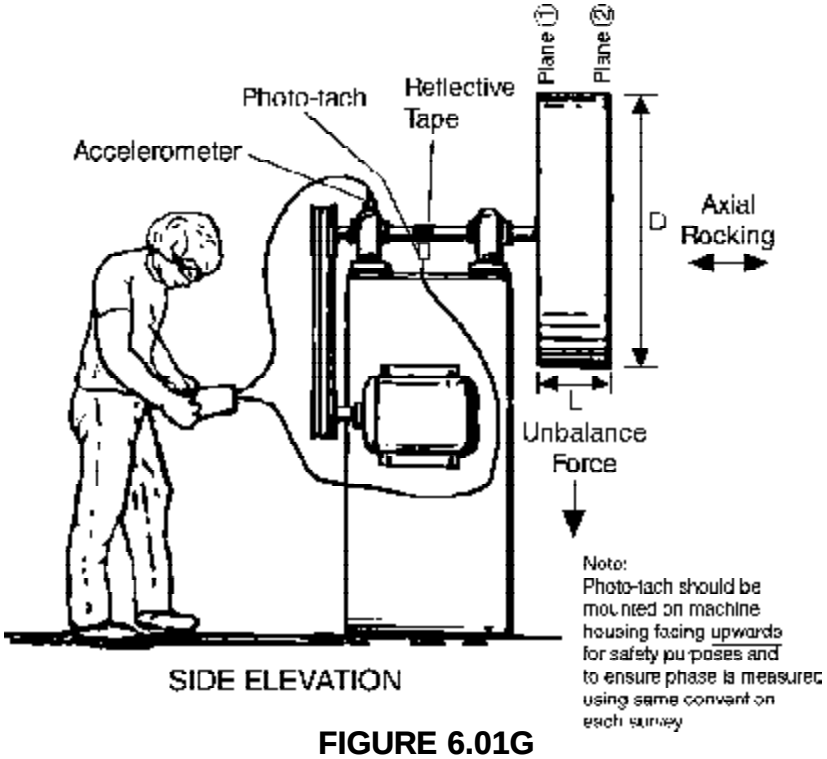


FIGURE 6.01G
FIELD INSTRUMENT SETUP FOR BALANCING OVERHUNG ROTORS

Therefore, either the data collector can be used using single-plane balance software or the single-plane graphic technique previously explained can be successfully employed on many overhung rotors, particularly if the ratio of the rotor length-to-diameter (L/D) is less than approximately .50 (where L is length of the rotating component on which correction weights will be placed and D is the diameter of this component - see Figure 6.01G).

Following below will be a description of this classic single-plane balancing technique for overhung rotors:

a. Set Up Data Collector and/or Spectrum Analyzer Instruments - The data collector, phototech, accelerometer and so forth should be set up as previously described under Section D and Figure 6.01G showing the two-plane balancing procedure. Alternatively, the analyst may wish to employ either a swept-filter analyzer which drives a strobe light (like an IRD 350 or IRD 880),or a spectrum analyzer which will fire a photo-tach for phase measurement.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

b. Take Initial Measurements - Take initial measurements of 1X RPM amplitude, frequency and phase before adding any trial weights. Measurements should be taken on both the outboard and inboard bearings in both vertical and horizontal directions. The radial direction measurement having the highest amplitude will normally be employed for initial balancing (however, after correcting unbalance in the radial direction, measurements will have to be taken in the other radial direction to ensure amplitudes in it are likewise acceptable).

c. Determine if the Dominant Problem is Either Static or Couple Unbalance - Looking at the amplitude and phase measurements taken on both bearings in the radial and horizontal directions, determine if the problem is dominated by either static or couple unbalance. If phase differences between the outboard and inboard bearing are approximately 140° or more in both the vertical and horizontal directions, the dominant problem will be couple unbalance. On the other hand, if these differences are both anywhere from 0° to approximately 40°, static unbalance is dominant. Of course, phase differences ranging from approximately 40° to 140° are truly dynamic balance once again with a combination of static and couple. If the problem appears to be mostly couple unbalance, use couple unbalance procedures outlined below. However, if the problem appears to be predominantly static or dynamic unbalance, employ static balance procedures. For now, we will assume that the problem is mostly static.

d. Make a Single-Plane Static Balance - Referring to Figure 6.01G, use single-plane techniques taking measurements on Bearing A and placing trial and correction weights in Plane 1.

e. Determine if Resultant Vibration Amplitudes Meet Required Criteria - After completing the single-plane static balance using Plane 1, repeat vibration measurements on both the outboard and inboard bearings in each direction (including axial) and ensure that amplitudes now meet allowable criteria.

f. If Considerable Couple Unbalance Now Remains, Continue With Single-Plane Balance From Bearing B - Overhung rotors often have large cross-effects which means that single-plane balancing from Plane 1 will often cause high vibration over at Bearing B. Therefore, the analyst will perform another single-plane balance, this time making his measurements from Bearing B farthest from the component to be balanced. When he arrives at the single-plane correction weight solution, he should place this weight in Plane 2; and then place an identical size correction weight over in Plane 1 some 180° away from the weight location in Plane 2.

g. Determine if Amplitudes Now Meet all Criteria - After completing the single-plane couple correction, the analyst must again make measurements in horizontal, vertical and axial directions on each bearing and determine that all amplitudes now meet allowable criteria. Often, further balancing must be done at this point beginning with another single-plane balance using Bearing A and Plane 1 which might possibly be followed by another couple balance correction.

h. If Allowable Criteria Cannot be Met in all Three Directions of Each Bearing, Proceed to Two-Plane Balance Procedure Outlined Below - Sometimes, this single plane approach will not successfully reduce amplitudes below allowable criteria in all three directions on each bearing, particularly if the L/D ratio is greater than .50 or if the component to be balanced is located far away from the closest bearing. If this happens, two-plane techniques outlined below will have to be taken.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

2. Balancing Overhung Rotors by Classic Two-Plane Static-Couple Method:

Due to the significant cross-effects which are often present in overhung rotors, two-plane balance correction techniques often are more successful than those employing single-plane methods. However, one of the problems with two-plane methods is that it can sometimes be a little confusing on deciding which bearing is the left and which is the right bearing; similarly, which plane is the left and which is the right plane? (Some data collectors refer to these as the near and far planes as opposed to left and right; terminology does not matter - only that the analyst remain consistent in his convention.) Referring to Figure 6.01G, when using two-plane techniques, Bearing A will be considered the bearing closest to the overhung rotor while Bearing B will be closest to the pulley. Similarly, Plane 1 will be on the inboard side of the wheel closest to the bearings whereas Plane 2 will be outboard.

Here again, a static/couple solution will be employed when the two-plane correction weight calculations are completed. Since most overhung rotors are so sensitive to static unbalance, only the static correction weight will be placed when this static/couple solution is obtained. Then, after trim balancing, if considerable couple unbalance remains, the analyst will proceed to correct this as well. He should follow the procedure outlined below:

a. Set Up Instruments as Outlined in Two-Plane Balance Method in Figure 6.01G - Here again, this same procedure can be used with either data collectors, swept-filter analyzers or real-time analyzers. However, if using either a swept-filter or real-time analyzer, the analyst should have a two-plane calculator program that is capable of providing static/couple solutions.

b. Take Initial Measurements on Both Bearings - Here again, 1X RPM amplitude, frequency and phase should be measured in horizontal, vertical and axial directions on both the outboard and inboard bearings.

c. Complete a Two-Plane Balancing Procedure, But Do Not Yet Place Balance Correction Weights - A two-plane balance procedure like that outlined in Section D should be employed, but final correction weights not put in place. Instead, when the trial weights sizes and locations are calculated for each plane, the analyst should ask for a static/ couple solution and should initially only make the static correction. For example, if the static solution called for 1 oz in Plane 1 whereas the couple solution called for a 2 oz correction in Planes 1 & 2 180° opposite one another, make only the static correction at this point.

d. Determine if Amplitudes Now Meet Allowable Criteria - After making the static correction in Plane 1, see if amplitudes in all three directions on each bearing are now within compliance with allowable criteria. If not, trim as required. Again, when the two-plane corrections are determined, ask for the static/couple solution and once again, make only the static correction. Most of the time, the problems are resolved at this point. However, if considerable couple unbalance still remains, complete another two-plane procedure again asking for the static/couple solution - this time making the couple correction called for, and not the static correction.

e. Determine if Amplitudes Now Meet Allowable Criteria - After each of the two trials making these static corrections and the single trial making the couple correction, compare amplitudes in horizontal, vertical and axial directions on both the outboard and inboard bearings with allowable criteria. A small percentage of the time, the couple correction will throw the static balance back off. If this is the case, it may require one more static correction before the rotor is successfully balanced.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

6.015 Allowable Residual Unbalance and ISO Balance Quality Grade:

When balancing a rotor, one needs to know to what precision he is required to balance the unit. In reality, it is not enough to simply say that it will be acceptable to balance the machine to a level of .10 in/sec or 1.0 mil at a certain speed. While one rotor may satisfactorily be balanced at such levels, another will not. This was recognized back in the 1950's by experts involved in the balance field. They recognized that the residual unbalance is truly proportional to the amount and radius of the remaining rotor eccentricity as well as the weight of the rotor itself and its operating speed. Therefore, they developed a series of balance tolerances known as ISO Standard No. 1940 on "Balance Quality of Rotating Rigid Bodies". Table 6.01A provides the balance quality grades as per these standards for a whole group of rotor types. Then, Table 6.01B provides the numerical standards for each of the ISO balance quality grades (ISO G-1, ISO G-2.5, ISO G-6.3, etc.). Note that the lower the G tolerance, the more precision the balance quality grade. Also note that it is based on the rotor RPM (horizontal axis) as well as the residual unbalance per pound of rotor weight (vertical axis). Table 6.01C is provided showing these same balance quality grades delineated by bands separating one balance quality grade from another. Table 6.01C also shows common nominal RPM's in the United States (1200, 1800 and 3600 RPM).

When balancing a machine, you first refer to the tabulated information in Table 6.01A to determine to what tolerance you should balance it. For example, if balancing an automobile crankshaft, this falls under ISO G-16 quality grade. On the other hand, fans fall under G-6.3, and grinding-machine drives fall under G-1. Note that the quality grade number itself represents the maximum permissable circular velocity of the rotor center of gravity expressed in millimeters per second (mm/sec). For example, a quality grade G-6.3 corresponds to rotor velocity of 6.3 mm/ sec RMS which corresponds to an equivalent .248 in/sec RMS (.351 in/sec peak).

It is the experience of the author that this ISO Standard, which was established in 1966, is a little too conservative, possibly because it might be based on the technology available during that day and time. It is recommended that when using the ISO tolerance, that you should use one quality grade better than that specified for the specific machine you wish to balance (that is, if the standard calls for G-6.3, we would recommend using ISO G-2.5). For example, if balancing a fan wheel, note that Table 6.01A calls for an ISO quality grade G-6.3 for fans. In this case, we would recommend ISO grade G-2.5. Following below will be a procedure on how to determine the allowable residual unbalance, the ISO balance quality grade that you have achieved, and the rotor balance sensitivity:

How to Determine Residual Unbalance Remaining in a Rotor After Balancing

When field balancing, one must know when to determine that the job is complete. He will know this not only when he has achieved low vibration levels, but also when he knows he has balanced the rotor within allowable specifications. To know this, he must determine the residual unbalance remaining in the rotor. This can be accomplished by following the procedure below (refer to Figure 6.01H):

a. Make original measurements of amplitude and phase and graph this to scale on polar coordinate paper. Call this vector the "O" vector.

b. Attach a trial weight and document the trial weight size (oz) and radius (in) to which it is attached. (mr = trial weight size X trial weight radius)

c. After attaching the trial weight, spin the rotor and measure amplitude and phase. Graph this on the polar coordinate paper as the "O + T" vector.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

d. Draw a vector called “T” from the end of vector “O” to the end of vector “O + T”. Vector "T" represents the effect of the trial weight alone. Measure the length of vector "T" to the same scale as that used for vectors "O" and "O + T". Using this scale, determine the equivalent vibration level (mils).

e. Calculate Rotor Sensitivity as per the following equation:

$$\text{Rotor Sensitivity (oz-in/mil)} = \frac{\text{(Trial Wt. Size)(Trial Wt. Radius)}}{\text{Trial Weight Effect}}$$

(Eqn. 6.01A)

f. Calculate Residual Unbalance using Equation 6.01B. If Residual Unbalance not brought within tolerances, trim balance using current correction weight as the trial weight for the trim run. Continue trim balancing until Residual Unbalance is reduced within required balance tolerances:

$$\text{Residual Unbalance (oz-in)} = \text{Rotor Sens. (oz-in/mil)} \times \text{Vib.Ampl. After Bal. (mils)}$$

(Eqn. 6.01B)

Example (see Figure 6.01H):

Given:

Required ISO Balance Quality = G 2.5
Rotor Weight = 100 lb
Rotor Speed = 800 RPM
Amplitude After Balancing = 2.0 mils

Therefore, Required U_{per} = 1.76 oz-in total (single-plane balance)

a. Original reading = 10 mils @ 240° = “O” vector.

b. Trial weight of 3 oz is attached in the balance plane at a 6 inch radius (mr = 3 oz X 6 in = 18 oz-in)

c. Trial run reading = 8 mils @ 120° = “O + T” vector

d. Effect of trial weight alone = T = 15.5 mils (from Figure 6.01H)

e. Rotor Sensitivity = $\frac{18 \text{ oz-in}}{15.5 \text{ mils}} = 1.16 \frac{\text{oz-in}}{\text{mil}}$

f. Residual Unbalance = $(1.16 \frac{\text{oz-in}}{\text{mil}})(2.0 \text{ mils}) = 2.32$ oz-in (not within specs)

Continued balancing and reduced vibration to 1.0 mil

$$\text{Residual Unbalance} = (1.16 \tfrac{\text{oz-in}}{\text{mil}})(1.0 \text{ mil}) = 1.16 \text{ oz-in (in compliance)}$$

© Copyright 2001 Technical Associates Of Charlotte, P.C.

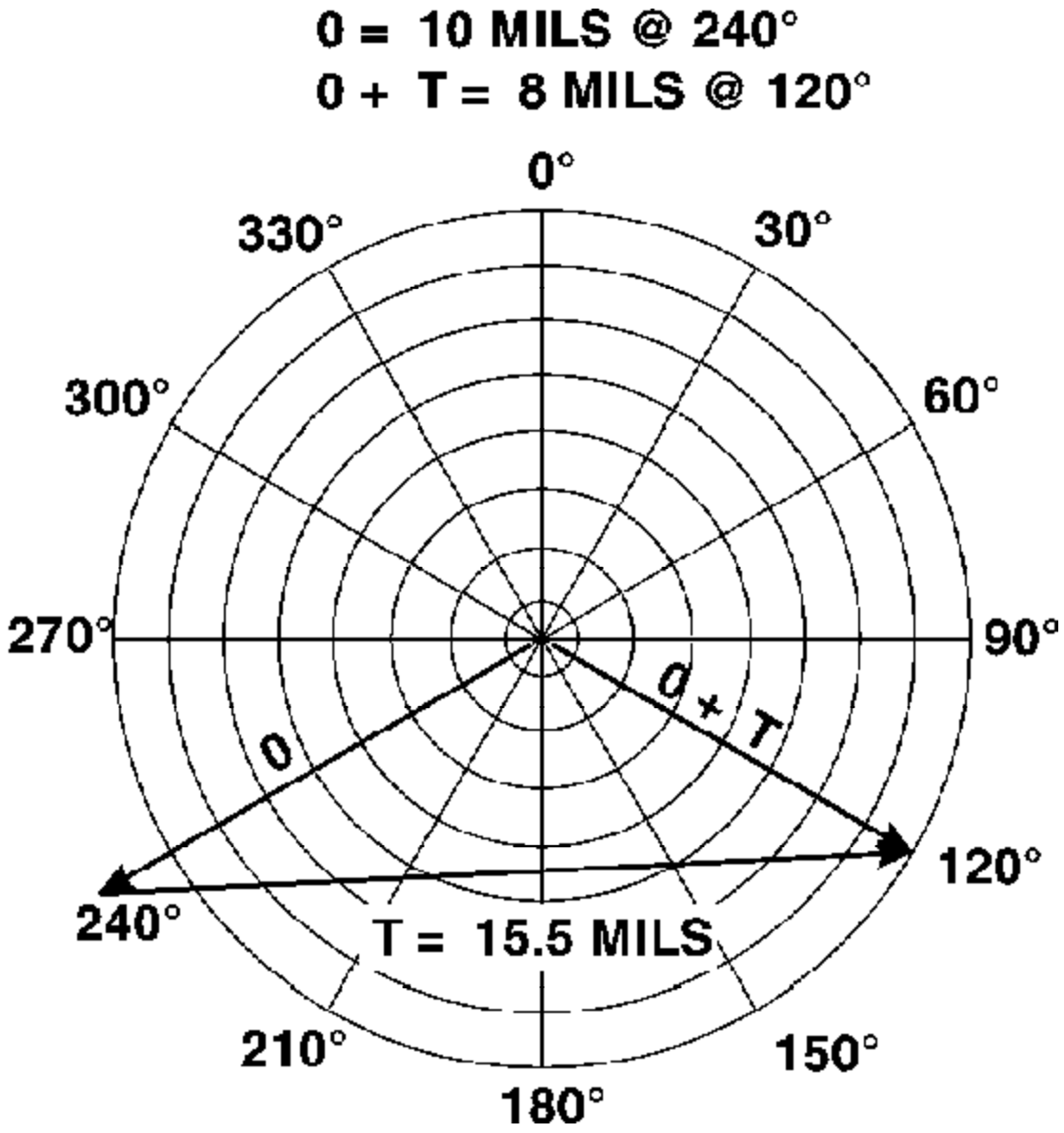


FIGURE 6.01H
STANDARD SINGLE-PLANE VECTOR SOLUTION

© Copyright 2001 Technical Associates Of Charlotte, P.C.

TABLE 6.01A*
BALANCE QUALITY GRADES FOR VARIOUS GROUPS OF REPRESENTATIVE RIGID ROTORS IN ACCORDANCE WITH ISO 1940 AND ANSI S2.19-1975
***(Reference 20)**

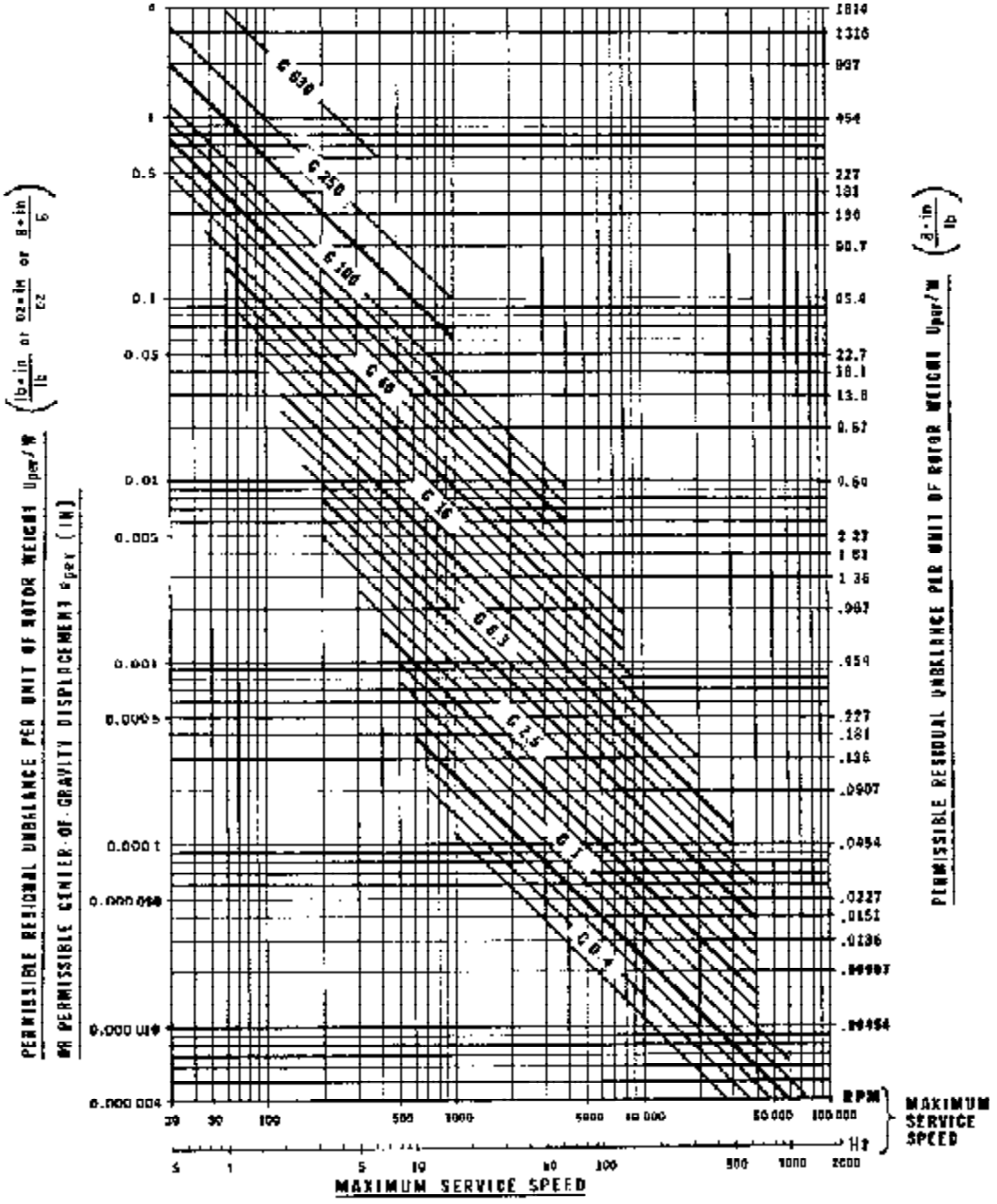
Balance Quality Grades G	EW 1, 2 (mm/sec)	Rotor Types - General Examples
G 4000	4000	Crankshaft drives[4] of rigidly mounted slow marine diesel engines with uneven number of cylinders.[6]
G 1600	1600	Crankshaft drives of rigidly mounted large two-cycle engines.
G 630	630	Crankshaft drives of rigidly mounted large four-cycle engines. Crankshaft drives of elastically mounted marine diesel engines.
G 250	250	Crankshaft drives of rigidly mounted fast four-cylinder diesel engines.[4]
G 100	100	Crankshaft drives of fast diesel engines with six or more cylinders.[4] Complete engines (gasoline or diesel) for cars, trucks and locomotives.[5]
G 40	40	Car wheels, wheel rims, wheel sets, drive shafts. Crankshaft drives or elastically mounted fast four-cycle engines (gasoline or diesel) with six or more cylinders.[4] Crankshaft drives for engines of cars, trucks or locomotives.
G 16	16	Drive shafts (propeller shafts, cardan shafts) with special requirements. Parts of crushing machinery. Parts of agricultural machinery. Individual components of engines (gasoline or diesel) for cars, trucks, and locomotives. Crankshaft drives of engines with six or more cylinders under special requirements. Slurry or dredge pump impeller.
G 6.3	6.3	Parts of process plant machines. Marine main turbine gears (merchant service). Centrifuge drums. Fans. Assembly aircraft gas turbine rotors. Fly wheels. Pump impellers. Machine-tool and general machinery parts. Normal electrical armatures. Individual components of engines under special requirements.
G 2.5	2.5	Gas and steam turbines, including marine main turbines (merchant service). Rigid turbo generator rotors. Rotors. Turbo-compressors. Machine-tool drives. Medium and large electrical armatures with special requirements. Small electrical armatures. Turbine-driven pumps.
G 1	1	Tape recorder and phonograph (gramophone) drives. Grinding machine drives. Small electrical armatures with special requirements.
G 0.4	0.4	Spindles, disks, and armatures of precision grinders. Gyroscopes.

1. The quality grade number represents the maximum permissible circular velocity of the center of gravity (CG) in mm/sec.
2. $\omega = 2\pi n/60 \approx n/10$, if n is measured in revolutions per minute and ω in radians per second.
3. In general, for rigid rotors with two correction planes, one-half of the recommended residual unbalance is to be taken for each plane; these values apply usually for any two arbitrarily chosen planes, but the state of unbalance may be improved upon at the bearings. For disk-shaped rotors the full recommended value holds for one plane.
4. A crankshaft-drive is an assembly which includes the crankshaft, a flywheel, clutch, pulley, vibration damper, rotating portion of connecting rod, etc.
5. For purposes of this Standard, slow diesel engines are those with a piston velocity of less than 9 mm/sec; fast diesel engines are those with a piston velocity of greater than 9 m/sec.
6. In complete engines, the rotor mass comprises the sum of all masses belonging to the crankshaft drive described in Note 3 above.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

TABLE 6.01B*
MAXIMUM PERMISSIBLE RESIDUAL SPECIFIC UNBALANCE CORRESPONDING TO VARIOUS BALANCE QUALITY GRADES G, IN ACCORDANCE WITH ISO 1940

***(Reference 20)**



NOTES:

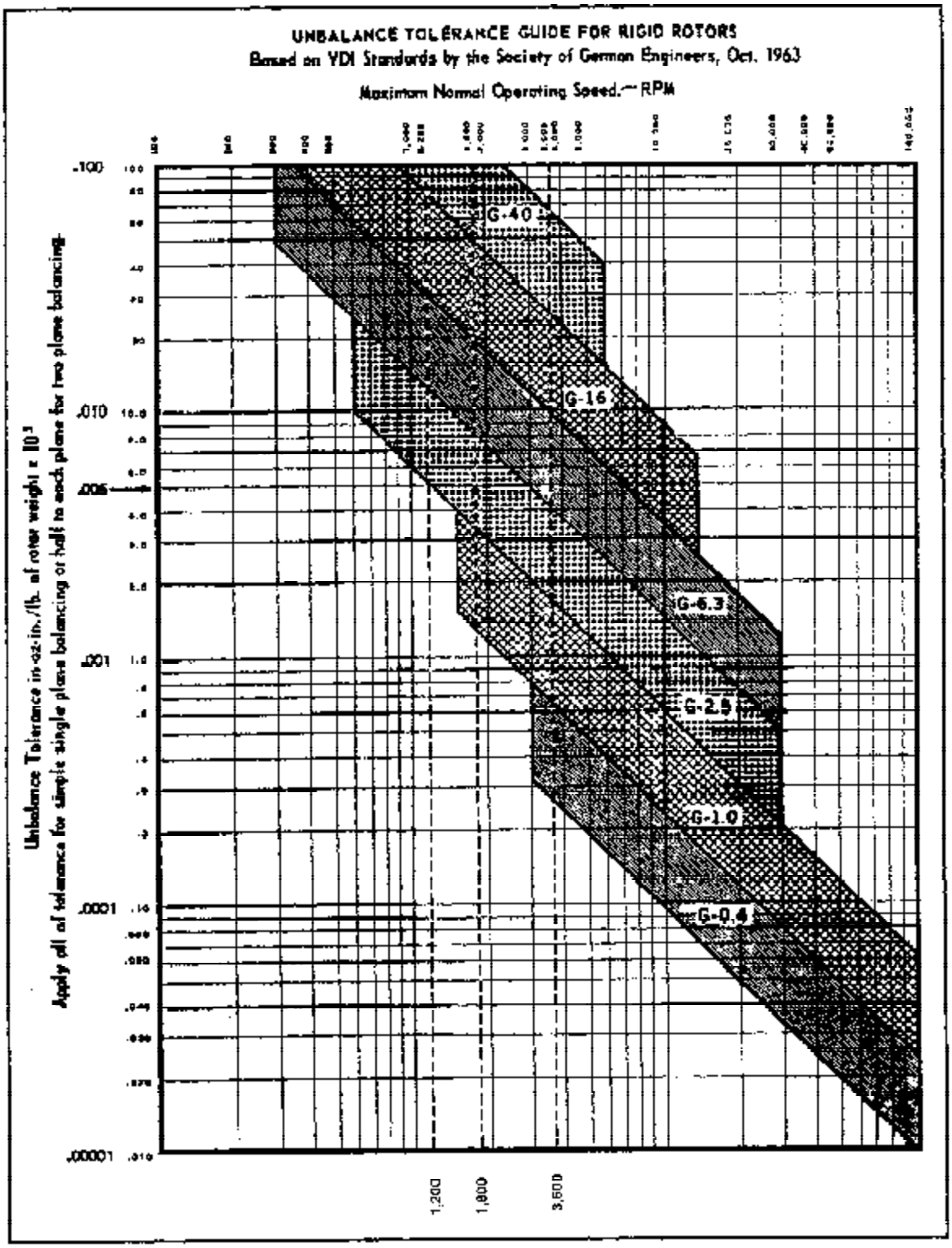
(1) In general, for rigid rotors with two correction planes, one half of the recommended residual unbalance is allowed for each plane; these values apply usually for any two arbitrarily chosen planes. (See paragraph on Applying Tolerances to Two-plane Rotors). For disc-shaped rotors the full recommended value applies to one plane.

(2) To obtain acceptable residual unbalance in oz-in per lb of rotor weight multiply lb-in value by 16.

(3) To avoid need for multiplication use appropriate Balance Tolerance Nomogram on following pages.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

TABLE 6.01C*
EQUIVALENT ISO QUALITY GRADES SHOWN IN BANDED REGIONS
***(Reference 21)**



© Copyright 2001 Technical Associates Of Charlotte, P.C.

6.02 ECCENTRIC ROTORS

McGraw Hill's "Dictionary of Mechanical and Design Engineering" defines Eccentricity as:

"the distance of the geometric center of a revolving body from the axis of rotation".

In other words, referring to Figures 6.02A through 6.02C, an eccentric rotor is one in which the shaft centerline does not line up with the rotor centerline. This results in more weight being on one side of the rotating centerline than the other and causes the shaft to wobble in an irregular orbit. This is inherently unstable and can be the source of troublesome vibration. Sometimes, it is possible to "balance out" part of the effect of eccentricity, but much of the displaced motion still remains. In other cases, it is not even possible to perform a good balance on rotors having more eccentricity. Today with the emphasis on higher and higher rotating speeds, it is very important that eccentricity be minimized.

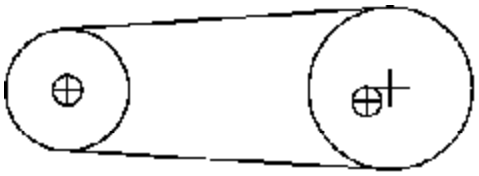


FIGURE 6.02A
ECCENTRIC SHEAVE

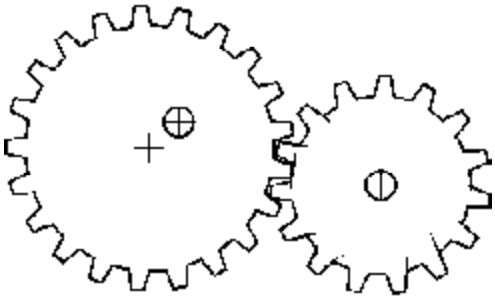


FIGURE 6.02B
ECCENTRIC GEAR

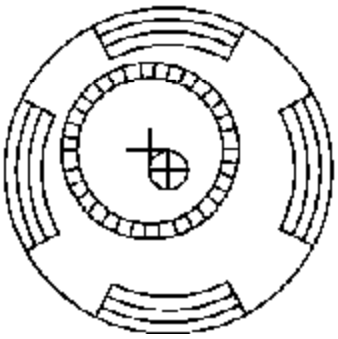


FIGURE 6.02C
ECCENTRIC MOTOR ARMATURE

© Copyright 2001 Technical Associates Of Charlotte, P.C.

The classic spectrum of an eccentric rotor is shown on Figure 6.02D. Note that, like unbalance, the spectrum is dominated by the frequency at 1X RPM of the eccentric component, particularly when the measurement is taken in the direction through the centers of the two rotors. Looking at Figure 6.02D which shows a motor driving a fan with an eccentric pulley, note that the motor operating speed peak will be much lower than that at fan speed, particularly when taken in line with the belt direction. This eccentricity causes a very highly directional loading so that vibration at 1X RPM can be very much higher in one radial direction than in the other (depending on the amount of eccentricity).

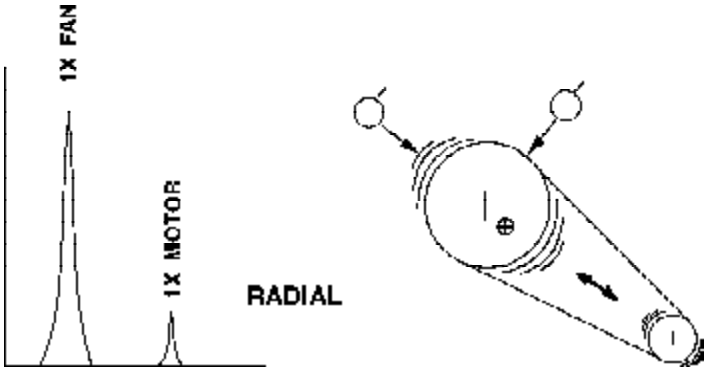


FIGURE 6.02D
TYPICAL SPECTRUM OF AN ECCENTRIC ROTOR

An eccentric rotor exhibits each of the following characteristics:

1. Some of the more common types of eccentric rotors include eccentric pulleys, gears, motor rotors and pump impellers:

    a. Figure 6.02A shows an eccentric pulley. In these units, the largest vibration most often occurs in the direction of belt tension and at the frequency of 1X RPM of the eccentric pulley. Eccentric pulleys represent one of the most troublesome sources of undesirable vibration in belt drives today. Unfortunately, the industry to date has not sufficiently policed itself to minimize eccentricity in common pulleys. Often, attempts are made to overcome pulley eccentricity after the fact by balancing. Even when this is done, balancing alone will not significantly lower the back-and-forth belt motion which results in continuous belt tension variation, depending on the position of the eccentric pulley at any instant. Plants need to protect themselves by writing eccentricity specifications into their belt drive orders if they want to maximize the life of their machinery and lower their vibration.

    b. Figure 6.02B shows an eccentric gear in which the largest vibration will occur in a direction in line with the centers of the two gears, and at a frequency of 1X RPM of the eccentric gear. The vibration signature will appear like unbalance of this gear, but it is not. If the eccentricity is significant, it can induce very high dynamic loads on gear teeth as they are forced into and out of a bind with the mating gear. Phase analysis can be used on gears having high 1X RPM vibration to differentiate whether unbalance or eccentricity is the source (See characteristic #3 below). Not only do eccentric gears result in higher 1X RPM vibration, but they also can generate high amplitude gear mesh frequencies and harmonics which will be accompanied by higher than normal amplitude sideband frequencies spaced around the gear mesh frequency at the eccentric gear RPM. Sometimes, these sidebands will be at 2X RPM of the eccentric gear. These sidebands will modulate the amplitude of gear mesh frequencies themselves.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

c. Figure 6.02C shows an eccentric motor rotor. Eccentric rotors produce a rotating variable air gap between the rotor and stator which induces pulsating vibration between 2X line frequency (7200 CPM) and its closest running speed harmonic as well as generating pole pass frequency (F_p) sidebands around 2X line frequency (see "Electrical Problem Vibration Symptoms" in Section 6.12). That is, for a 3580 RPM motor, this would be between 2X running speed and 2X line frequency, whereas for a 1780 RPM unit, it would be between 4X RPM and 7200 CPM Section 6.12 will show that an eccentric motor rotor will also generate pole pass frequency sidebands around 2X line frequency (where pole pass frequency, F_p, equals #Poles times slip frequency). Finally, the eccentric rotor motion itself will cause a variation in the magnetic field between the stator poles and rotor, thereby inducing 1X RPM vibration between the rotor and stator.

d. Eccentric pump impellers can result in unequal hydraulic forces distributed between the rotating impeller and stationary diffuser vanes. This can result not only in high vibration at pump RPM, but also at vane pass frequency and multiples (# vanes times RPM and multiples) due to a "hydraulic unbalance" induced by the eccentric impeller.

2. Attempts to balance eccentric rotors will often result in reducing vibration in one direction, but increasing it in the other radial direction.

3. Eccentric rotors may cause significantly higher vibration in one radial direction than in the other (as does resonance, wiped bearings and sometimes looseness as well). Phase analysis can be employed as an effective tool to detect whether or not the source of high vibration at 1X RPM is from eccentricity or from another 1X RPM source such as unbalance. Comparative horizontal and vertical phases usually differ by approximately 0° or 180° since the force induced by eccentricity is highly directional (rather than a 90° phase difference in horizontal and vertical as in the case of dominant unbalance problems).

© Copyright 2001 Technical Associates Of Charlotte, P.C.

6.03 BENT SHAFT

A bowed or bent shaft can generate excessive vibration in a machine, depending on the amount and location of the bend. Like eccentric shafts, the effects can sometimes be decreased by balancing. However, more often than not, it is not possible to achieve a satisfactory balance in a shaft which has any noticeable bend. Analysts are sometimes successful in removing the bend by various techniques sometimes involving thermal treatments. In these cases, however, one must be careful not to introduce residual stresses which might later lead to shaft fatigue. Bent shafts exhibit the following characteristics:

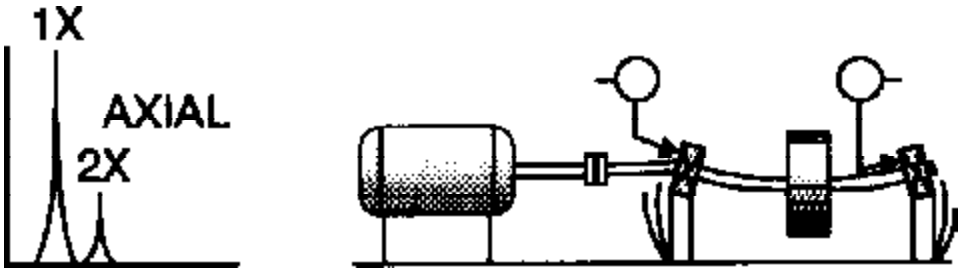


FIGURE 6.03A
BENT SHAFT SPECTRAL AND PHASE RESPONSE

1. Figure 1 shows that high axial vibration is generated by the rocking motion induced by the bent shaft. Dominant vibration normally is at 1X RPM if bent near the shaft center, but a higher than normal 2X RPM component can also be produced, particularly if bent near the coupling.

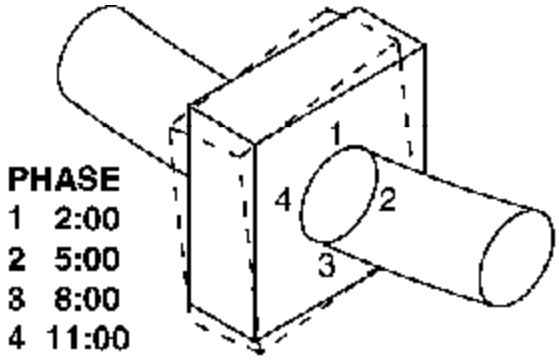
2. Axial phase change between two bearings on the same component (motor, fan, pump, etc) approaches 180°, dependent on the amount of the bend (as shown in Figure 1). In addition, if one makes several measurements on the same bearing at various points in the axial direction, he will normally find that phase differences approaching 180° occur between that measured on the left and right hand side of the bearing, and also between the upper and lower sides of the same bearing.

3. Amplitudes of 1X RPM and 2X RPM will normally be steady, assuming that 2X RPM is not located close to twice line frequency (7200 CPM) which might induce a beat of the 2X RPM component with 2X line frequency if there is high electromagnetic vibration present.

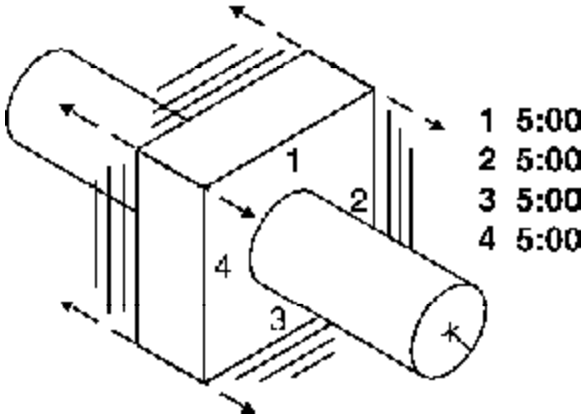
4. Please note the axial phase measurements on 4 points of a bearing housing pictured in Figure 6.03B. If the shaft is bowed through or very near a bearing, you get a twisting motion by the bearing housing itself which will result in significantly different phase readings on this bearing housing in the axial direction as pictured in Drawing A of Figure 6.03B. Drawing B of this figure shows the axial phase which results from a true, straight shaft.

5. When much run-out is present at the rotating mass, it appears as unbalance. When run-out at the coupling occurs, it appears as misalignment.

© Copyright 2001 Technical Associates Of Charlotte, P.C.



A: AXIAL PHASE MEASUREMENTS INDICATING TWISTING MOTION DUE TO A BENT SHAFT



B: AXIAL PHASE MEASUREMENTS INDICATING A TRUE SHAFT IN PROPER MOTION

FIGURE 6.03B

6. In bent shafts, amplitude can vary with the square of speed and preload. If unbalance is more of the problem than bow, vibration will decrease abruptly if operating below the first critical speed. However, if the rotor is brought above its first critical speed, unbalance amplitude will change only a small amount, whereas if the dominant problem is a bent shaft, the amplitude will again drop significantly as the speed is dropped towards the first critical speed.

7. If a rotor is located between bearings and should operate at or close to its fundamental natural frequency, it will appear to be a "bent" shaft and will display these symptoms (see Figure 6.05E in Section 6.05 on "Resonant Vibration"). However, this is only temporary. When the machine is stopped or at another non-resonant speed, it will then "straighten out".

8. When electric motors have problems such as shorted laminations, they will thermally induce a bend as the machine heats up, with the resultant vibration getting higher and higher as the rotor heats. This again will introduce bent shaft symptoms (see Figure 6.09H in Section 6.09 on "Electrical Vibration"). In this case, the shaft again will straighten when allowed to come back to room temperature if the plastic limit of the shaft material has not been exceeded. This will be covered later in the electrical problems Section 6.09.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

6.04 MISALIGNMENT

Likely the most widespread mechanical problem in industry today is misalignment. Many plants have begun to resolve a number of their unbalance problems as more and more data collectors having this capability, as well as other analyzers are acquired by plants, and balance actions are taken. However, new instruments are also now becoming available to resolve alignment problems. These include optical as well as the newer laser devices. With these newer instruments, we can now realize that machines have been operating for quite some time with much higher levels of misalignment than had previously been thought. In fact, we are learning that it is not uncommon at all to have 30% to 50% or more of machines in any plant that have high degrees of misalignment.

The trouble with such high levels of misalignment is that it induces high vibration levels leading to premature failure of expensive machine components and increased energy demands as well. Misalignment is now probably one of the leading causes of bearing failures as well.

Although vibration responds to the degree of misalignment, there is not a direct 1-for-1 relationship between the amount of misalignment (angularity and offset) and the amount of vibration. As John Mitchell states on page 182 of Reference 2:

> *"the vibration characteristics associated with misaligned flexible couplings are not a direct measure of the amount of misalignment but of the coupled system's ability to accommodate misalignment. Thus, the external symptoms of misalignment, in addition to being a function of the offset between shafts, are also affected by speed, torque, or any other condition such as corrosion or sludging which may alter the coupling's stiffness and hence its ability to accommodate a given offset."*

The first page of the Vibration Diagnostic Chart (Table 6.0) shows there are 3 types of alignment concerns including angular misalignment, parallel misalignment and a misaligned bearing cocked on a shaft. Each of these will be covered separately along with a section on coupling problems later. First, since misalignment problems are so prevalent today, a number of key facts should be considered about it including what effect it has on component lives; where it directs its potentially harmful forces; what are its spectral characteristics (harmonic content); what are its directional characteristics; what are its phase characteristics; and finally, what should be done to monitor alignment:

1. Component Failures Due To Misalignment - Misalignment can of course cause the coupling to fail, but other machine components as well. For example, if the coupling is stronger than the adjacent bearing, it can subject the bearing to excessive forces with little or no damage to the coupling. Similarly, such misalignment can detrimentally affect other components including gears, belts, sheaves, blading, etc.

2. Reaction On Free (or Outboard) End - It is possible for the highest reaction to misalignment to occur not on the bearing closest to the coupling, but on the free or outboard machine end. In these cases, incoming forces from the coupling may be strong enough to stabilize this system adjacent to the coupling and suppress the symptoms on this end.

3. Axial Vibration - Misalignment normally causes both high axial and radial vibration (as opposed to unbalance which acts mostly in the radial direction with the exception of overhung rotors).

© Copyright 2001 Technical Associates Of Charlotte, P.C.

4. Other Sources Of High Axial Vibration - Again, while misalignment is probably the most common source of high axial vibration, there are several other sources which can generate it including:
    a. Bent Shafts;
    b. Shafts in Resonant Whirl;
    c. Misaligned Bearings Cocked on the Shaft (See Section 6.043);
    d. Resonance of Some Component in the Axial Direction;
    e. Worn Thrust Bearings;
    f. Worn Helical or Bevel Gears;
    g. A Sleeve Bearing Motor Hunting for its Magnetic Center;
    h. Couple Component of a Dynamic Unbalance.

    Therefore, when high axial vibration occurs, do not quickly jump to the conclusion that the problem is misalignment. Instead, refer especially to phase; and then to the vibration spectrum.

5. Low Axial Vibration During Misalignment - Although misalignment is classically categorized as having high axial vibration, it does not always occur. For example, the writer has experienced some cases where misalignment was the problem even though axial levels were only about 1/4 of those in the radial direction. This is quite possible for machines with predominately parallel offset versus angular misalignment.

6. Comparable Horizontal And Vertical Amplitudes - Since it has been pointed out that it is possible for a machine to have good horizontal alignment but poor vertical, it is quite possible for misaligned machines to have much higher vibration in one radial direction versus another.

7. Radial Vibration Response To Misalignment - One would think that if driver and driven shafts were horizontally offset, it would cause high horizontal forces. Although this is sometimes the case, Reference 4 states that in most cases, high horizontal amplitudes are primarily the result of vertical misalignment and vice versa.

8. 2X RPM Vibration - Often, misalignment generates a higher than normal 2X RPM vibration which can act not only in the axial direction, but also in the radial. This second operating speed harmonic is caused by asymmetric stiffness in the machine and its supports, or in the coupling. That is, there is often quite a difference in stiffness around the supporting housing, frame, foundation and coupling itself which can allow a “back-and-forth” motion with each revolution, thereby resulting in 2X RPM vibration.

9. Higher Harmonics - Misalignment can also cause large numbers of harmonics which will make the spectrum appear like looseness/excessive clearance problems. The key distinguishing feature still appears to be the high level at 2X RPM in the axial direction. Several tests have been conducted purposely misaligning units and measuring their response (Reference 3). During these referenced tests, multiple harmonics often begin to appear when the misalignment became more and more severe.

10. Phase Is Best Indicator - When high vibration occurs on a machine predominately at 1X RPM and 2X RPM, the best overall indicator of misalignment problems is phase (that is, how the machine is shaking). Phase will differentiate between a number of other potential 1X RPM and 2X RPM vibration sources. Phase behavior in response to misalignment can be summarized as follows:

    a. Probably the best indicator of misalignment problems is evaluation of phase across the coupling. Here, one is checking how the driver shaft and its coupling half is

© Copyright 2001 Technical Associates Of Charlotte, P.C.

reacting relative to the driven shaft with its coupling half. When this phase difference across the coupling approaches 180° (±40° to 50°), misalignment is often indicated, particularly when other misalignment symptoms are present. The higher the severity of misalignment, the closer this difference will approach 180°. Similarly, the less significant are other problems such as unbalance, eccentricity, resonance, etc., the more this difference will approach 180°. Here, it is important that not only vibration spectra, but also phase measurements be captured in horizontal, vertical and axial directions on each of the bearing housings which are accessible.

b. Since it is possible for shafts to have good horizontal alignment, but poor vertical alignment (or vice versa), it is common in these cases for the horizontal phase difference to be quite different from the vertical phase difference. In fact, this is the case most of the time. In the special case where shafts have good horizontal, but poor vertical alignment, the shaft orbit itself would tend to be greatly elliptical which may make the alignment problem to appear to be eccentricity, resonance or a similar problem. That is, the amplitudes may be greatly different in one radial direction versus another. However, examination of the phase differences throughout the machine will indicate the misalignment problem.

c. When examining the phase difference on one of the rotors (just the motor, pump, fan, etc), the radial phase differences for significant misalignment will be either 0° or 180° (±30°). This is unlike unbalance in which such phase differences could be most anything (i.e., both the horizontal and vertical phase differences might be 70°). The key here is that misalignment phase differences will approach either 0° or 180°.

d. When comparing horizontal phase differences with vertical phase differences on the same rotor, about 90% of misaligned machines will show a difference approaching 180° between the vertical and horizontal. For example, if the horizontal phase differed about 30° between the outboard and inboard bearings, the vertical phase difference would be about 210° for most misaligned rotors. An unbalanced rotor will not show this phase behavior since whatever phase difference occurs on the horizontal direction will be very close to that in the vertical direction.

In summary, phase data should always be taken if possible on machines having high vibration at 1X and 2X RPM since phase will be the key indicator in differentiating whether the dominant problem source is misalignment as opposed to other problems of similar symptoms. While other symptoms such as high axial vibration and harmonic vibration are also good symptoms, these should not receive as much weight as phase (for example, if phase does indicate misalignment, but axial vibration does not, one should give more weight to the data provided him by phase) if the vibration is high.

11. Effect Of Other Problem Sources - When other problems such as unbalance, bent shaft, resonance, etc. are present, along with misalignment, this can affect not only the vibration spectrum, but also phase behavior. For example, if both unbalance and misalignment are present, it might show high levels at both 1X RPM and 2X RPM, plus radial phase differences which may or may not approach 150° to 180°, depending on the severity of each problem (in this case, axial phase differences across the coupling will still likely approach 180°. Reference 4 suggests that when several problem sources are all present, each of them will contribute vectorially. That is, if one had polar coordinate graph paper, you might show the contribution of unbalance as a 3 mil level at 30°; the misalignment a 2 mil level at 60°; and a simultaneous eccentricity problem contributing 1 mil at 0°. The resultant vector would not show a phase at any of these 3 individual angles, but instead would produce a vector somewhere on the order of 4 mils at about 40°. This would still not be radically different in any of the original phase angles.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

However, if each of these 3 were at widely differing phase angles, the resultant phase angle and magnitude could be quite different. In this case, one would first have to solve one problem at a time (in this case, he should likely solve first the eccentricity problem; then the misalignment, and finally balance the rotor). As each problem is solved, the vibration spectra and phase will react accordingly.

12. Monitoring Alignment Change - When aligning especially critical machines, it is often helpful to monitor the alignment and see how it might change. When doing so, it is especially important to monitor phase in all 3 directions at each bearing on each machine involved in the alignment. In most machines that have thermal offsets, if the machine is brought up to speed from room temperature, it should display misalignment symptoms in the beginning which should disappear as the machine comes up to full operating temperature. For example, the phase difference across the coupling should initially be on the order of 150° to 180°, but should drop to close to 0° to 30° in the end. In addition to phase, one should monitor how the vibration spectrum changes as well as other non-vibration related variables such as bearing temperature, temperature of the support legs and oil film pressure. When monitoring alignment change, each of the following should be considered:

    a. 1X RPM - It might be better to monitor vibration at higher harmonics of 2X up to 4X RPM rather than 1X RPM since the first harmonic will be effected by so many other things (unbalance, resonance, eccentricity, bent shaft, etc).

    b. 2X RPM - The 2X component should be a much better indicator of alignment than 1X assuming that this is not a 3600 RPM nominal motor (if so, 2X RPM will likely be very close to 2X line frequency which will "contaminate" the apparent 2X amplitude unless one is able to separate 2X RPM from 2X line frequency).

    c. 3X RPM - This 3X component may be the best indicator of alignment change if this particular machine does in fact cause an increase at 3X RPM with an alignment change. This is often the case. In such cases, it is not necessary for the 3X component to be larger than either 1X or 2X RPM, just that it be sensitive itself to alignment change.

    d. 4X RPM - The same analogy applies to 4X RPM as does 3X RPM with the exception being that this is not an 1800 RPM nominal speed machine (in which case 4X RPM would closely approach 2X line frequency at 7200 CPM).

    e. Number of Coupling Grids (or Segments) X RPM - Some coupling types include a number of grids or segments which often cause vibration at the number of grids (or segments) on one coupling half times the RPM, particularly when misalignment becomes severe. In these cases, where the coupling components themselves are effected and do respond to misalignment, this frequency will be an excellent choice to monitor since it will be well removed from any effects of unbalance, bent shaft, eccentricity or any other such source other than alignment.

6.041 Angular Misalignment:

Angular misalignment is pictured in Figure 6.04A. Each of the following characteristics are demonstrated by angular misalignment:

1. Angular misalignment primarily generates high axial vibration, particularly at 1X and 2X RPM. However, it is not unusual for either one of these peaks (1X, 2X or 3X RPM) to dominate alone.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

2. However typically, when the amplitude of either 2X RPM or 3X RPM exceeds approximately 30% to 50% of that at 1X RPM in the axial direction, angular misalignment is indicated. This assumes that there is high vibration (that is, misalignment may be of concern if 1X =.30 ips and 2X =.20 ips; but not if 1X = .03 and 2X = .02 ips).

3. Angular misalignment is best detected by 180° phase change across the coupling in the axial direction as pictured in Figure 6.04A. If each of the bearings on one of the side are moving one way, while those on the other side are moving in the opposite direction, angular alignment is highly suspect.

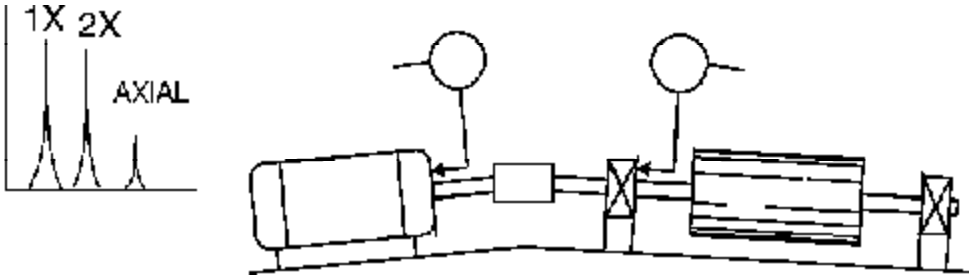


FIGURE 6.04A
ANGULAR MISALIGNMENT SPECTRAL AND PHASE RESPONSE

6.042 Parallel Misalignment (Also known as Radial Offset Misalignment):

Radial misalignment is pictured in Figure 6.04B. It displays each of the following characteristics:

1. Parallel misalignment primarily affects radial vibration as opposed to angular which affects axial.

2. Like angular alignment problems, parallel misalignment causes phase to approach 180° difference across the coupling, but in the radial direction (horizontal or vertical).

3. Radial misalignment is often indicated in a spectrum when 2X RPM exceeds approximately 50% of the amplitude at 1X RPM, but its height relative to 1X RPM is often dictated by the coupling type and construction. It is not uncommon for 2X RPM to exceed that at 1X RPM, particularly when the parallel misalignment becomes severe.

4. When either angular or parallel misalignment becomes severe, each can generate an array of harmonics ranging up to and including the 4th through the 8th harmonic. In this case, the severe misalignment spectrum can appear to be mechanical looseness (see page 1 of Table 6.0 Diagnostics Chart).

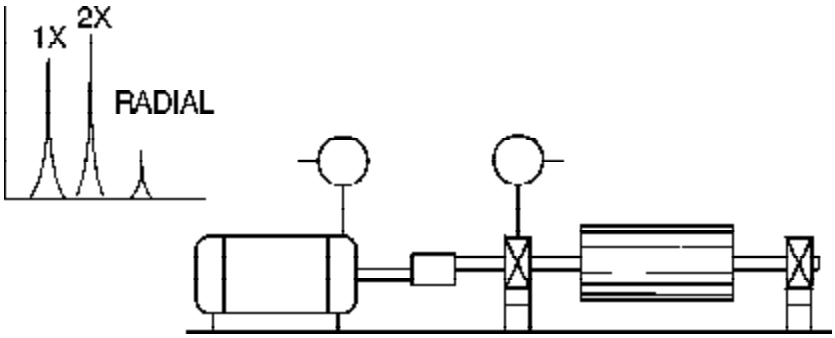


FIGURE 6.04B
PARALLEL MISALIGNMENT SPECTRAL AND PHASE RESPONSE

© Copyright 2001 Technical Associates Of Charlotte, P.C.

6.043 Misaligned Bearing Cocked on the Shaft:

When either a sleeve or rolling element bearing is misaligned or cocked on the shaft, it can cause high vibration and unusual loading. If it is detected, it should quickly be resolved before causing premature component failures. This problem is pictured in Figure 6.04C. Each of the following characteristics are indicative of a misaligned bearing on a shaft:

1. A cocked bearing will normally generate considerable axial vibration which can affect not only that at 1X RPM, but also 2X RPM as well.

2. If phase is measured in the axial direction at each of 4 points 90° apart from each other as shown in Figure 6.04D, a cocked bearing will be indicated by a 180° phase shift from top to bottom or from side to side.

3. Attempts to align the coupling or balance the rotor will not alleviate the problem. The effected bearing must be removed and correctly installed.

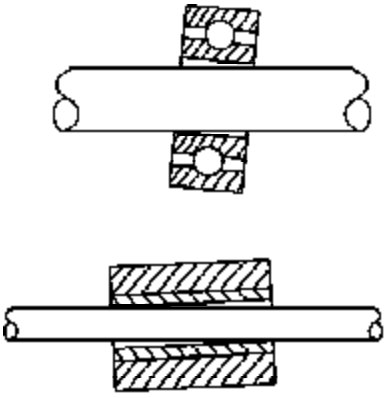


FIGURE 6.04C
MISALIGNED BEARINGS COCKED ON SHAFT

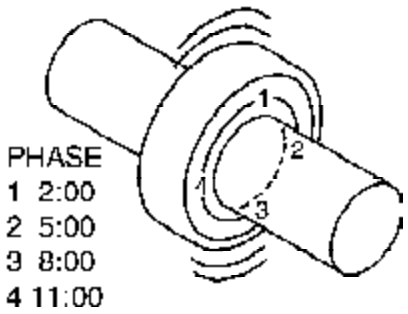


FIGURE 6.04D
AXIAL PHASE MEASUREMENTS INDICATING A COCKED BEARING ON A SHAFT

6.044 Coupling Problems:

It is often difficult to tell from vibration signatures or phase analysis whether the problem is misalignment or a coupling problem. Each of the many types of couplings has a different effect on the response of the machinery to which it is coupled. Other factors affecting its response include spacing between shafts, shaft diameter and bearing type. However, problem couplings do display the following characteristics:

© Copyright 2001 Technical Associates Of Charlotte, P.C.

1. 3X RPM will often respond to a coupling problem, particularly for a coupling having a spacer that is too long or too short. In these cases, the radial spectrum will normally indicate a fairly noticeable 3X running speed vibration, and that in the axial direction will show a much higher 3X RPM component. These problems are resolved by either correctly sizing the spacer or repositioning either the driver or driven equipment.

2. Gear Type Couplings can experience coupling lockup where the frictional force developed at gear teeth is greater than the applied force causing the coupling to become a rigid member. Friction welding of teeth can occur at this point, particularly if there is a lack of lubrication. A locked coupling can cause severe problems and may lead to thrust bearing failure if it results in the thrust load of 2 machines being applied to only one thrust bearing. Also, if the teeth do weld together and then break loose, it leaves pit marks on the coupling teeth. Coupling lockup can be broken temporarily either due to a change in load or by striking the coupling with a mallet or a piece of wood. However, this coupling should be closely inspected as soon as possible looking for tooth damage, lubrication problems and alignment problems, replacing if necessary.

   Coupling lockup will normally cause an increase in both axial and radial vibration with axial vibration normally being higher. Most of the time, the 1X RPM is most effected. However, certain types of couplings will generate a frequency distribution resembling a "Christmas Tree" effect. In these cases, many harmonics can appear with the vibration dropping approximately 25% from one harmonic to the next (Reference 5). What gives the spectrum a "Christmas Tree" effect is that there is a fairly uniform drop of about 25% all the way from the 2nd through the 5th or 6th harmonic.

3. A loose coupling is likely to cause sidebands around blade pass frequencies (#blades X RPM) and mesh frequencies (#teeth X RPM) as shown in Figure 6.04E (however, side-banding of blade pass and mesh frequencies does not always indicate a loose coupling). This is caused by the fact that a loose coupling does not drive the rotating equipment at a uniform speed, but rather, pulses at multiples of the shaft speed causing its running speed to modulate these other frequencies. Therefore, a signal similar to that shown in Figure 6.04E with equally spaced sidebands at coupling RPM can mean the coupling is loose (either from a poor fit on the shaft or from worn coupling components).

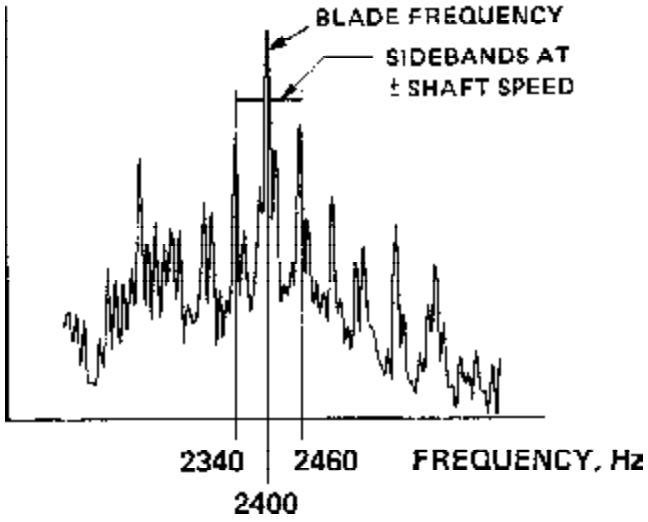


FIGURE 6.04E
LOOSENESS OF COUPLING INDICATED BY COUPLING RUNNING SPEED SIDEBANDS ABOUT BLADE PASS FREQUENCY

© Copyright 2001 Technical Associates Of Charlotte, P.C.

6.05 NATURAL FREQUENCIES AND RESONANCE

Natural frequencies and resonance are concepts that must be covered in any type of vibration course. The two are related; however, natural frequencies exist for every object in the universe while resonance is a special condition where some outside force is exciting a natural frequency. Natural frequencies, by themselves, are not harmful to a machine. Resonance, which does not always occur, can be extremely harmful to an object, such as a machine or support structure, and can determine the rate of failure of a machine's components. Also, a resonant condition can affect the performance of machinery and lower product quality. More often than not, natural frequency and resonance are discussed with an emphasis on mathematical theory rather than from a practical application viewpoint. This section will address these concepts as they pertain to condition monitoring.

6.051 Natural Frequency

Definition

The SHOCK AND VIBRATION HANDBOOK[1] defines Natural Frequency as "the frequency of free vibration of a system". In other words, when an object is standing still and is impacted with one short burst of energy, the object will vibrate at a series of frequencies which are known as its natural frequencies. Every object has an infinite number of natural frequencies and each of these natural frequencies has a corresponding physical shape associated with it called a "mode shape". An example of the first few natural frequencies and corresponding mode shapes for a shaft supported by bearings on each end (pinned-pinned beam) are given in **Figure 6.05A. Figure 6.05A** only shows the first 3 modes. In reality, there are an infinite number of natural frequencies and mode shapes, each higher in frequency than the last and with one more "hump" than the last. Note from the graphical representation of the mode shapes that the higher modes have less deflection from the shaft's position at rest. Therefore, less stress is typically induced in the shaft and bearings if a higher mode of vibration is excited.

Directionality

Not only does the shaft in **Figure 6.05A** have an infinite number of natural frequencies in the direction shown (in the plane of the page), it also has an infinite number of natural frequencies in the direction perpendicular to those shown in its horizontal direction (into and out of the page), as well as along the direction of the shaft in its axial direction (shortening and elongating the shaft). Each of these natural frequencies is unique (i.e. - they all occur at different frequencies).

Identification

The natural frequencies of an object can be identified by a variety of methods that are taught in later courses. However, a brief description of each will be given in this section.

1) IMPULSE NATURAL FREQUENCY TEST - An impulse natural frequency test is conducted by beginning with the machine at rest. An accelerometer is placed on a component of the machine to determine its natural frequencies. The machine is then impacted in the same or opposite direction while a FFT device (Data Collector or RTA) records the response of the accelerometer. Peaks that show up on the frequency spectrum are probable natural frequencies in the direction of the accelerometer. The test is repeated at many different locations and in all three spatial directions (horizontal, vertical and axial) to detect as many natural frequencies as possible.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

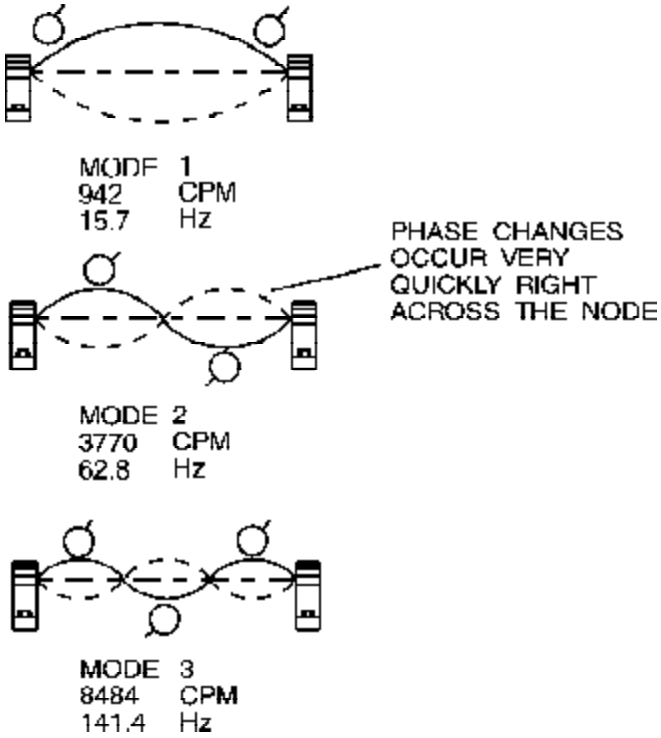


FIGURE 6.05A
MODE SHAPES DURING RESONANCE OF A SIMPLY SUPPORTED BEAM (SHAFT BETWEEN TWO BEARINGS)

2) COASTDOWN or RUNUP TEST - A coastdown test is conducted by placing the accelerometer on a machine component while the machine is running and then turning off the machine and letting it coast to a stop. An FFT device (Data Collector or RTA) records a series of spectra as the machine coasts down. Peaks will occur in the spectrum at suspected natural frequencies when the 1X RPM (unbalance force) excites them. Such natural frequencies can also be excited by other forcing frequencies as the machine coasts to a stop (2X RPM, blade pass frequency, gear mesh frequency, etc.). It is important to remember that a peak in the spectrum does not necessarily mean that a natural frequency exists. It could be that the speed of the rotor at that time equals the speed of another machine running nearby and a “beat” is formed which increases the amplitude at that particular frequency. To verify that a natural frequency is encountered, a Bode' plot can be run. A Bode' plot is similar to a coastdown plot except that phase is measured as well as the amplitude. A natural frequency is identified by a 90 degree phase shift as well as an increase in the amplitude. As the machine continues to coastdown, the phase will shift another 90 degrees for a total 180° phase shift when it passes through resonance and the amplitude will drop. **Figure 6.05B** is a plot of a coastdown and **Figure 6.05C** is a Bode' plot of the same coastdown. Note that the peak to the right is not a natural frequency since the Bode' plot does not show a phase shift. The peak to the left is a natural frequency, since the Bode' plot exhibits the 90 degree phase change at the natural frequency, and then continues onto its full 180 degree phase change.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

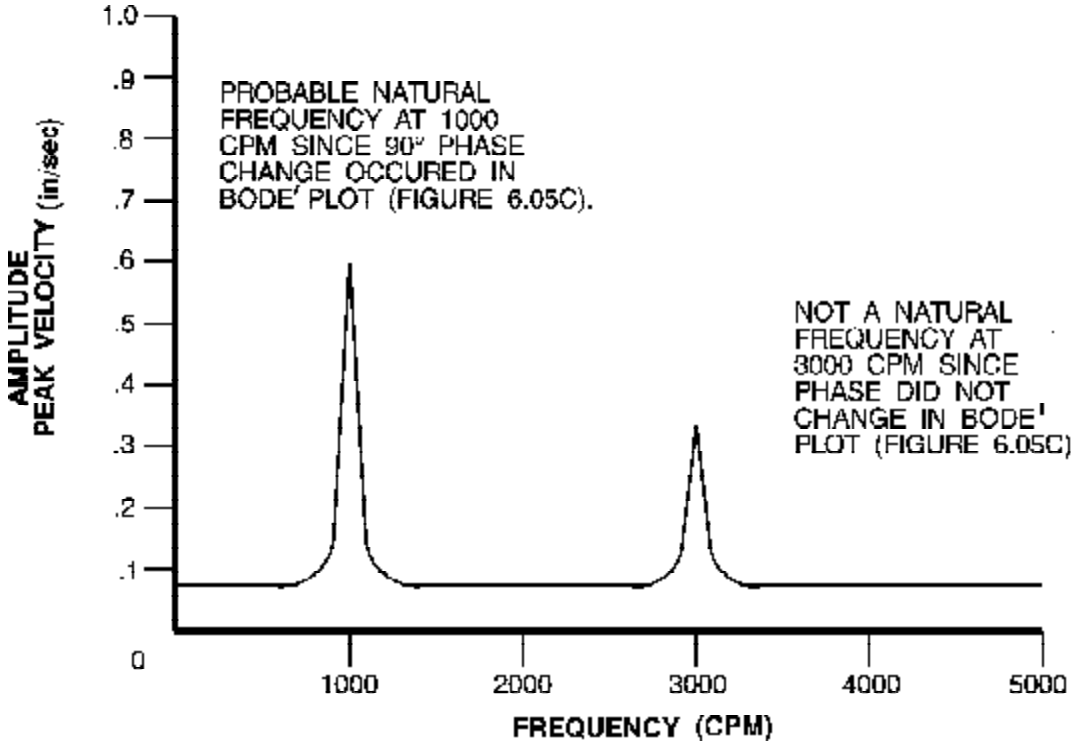


FIGURE 6.05B
COASTDOWN PLOT OF A MACHINE
(BODE' PLOT IS SHOWN IN FIGURE 6.05C)

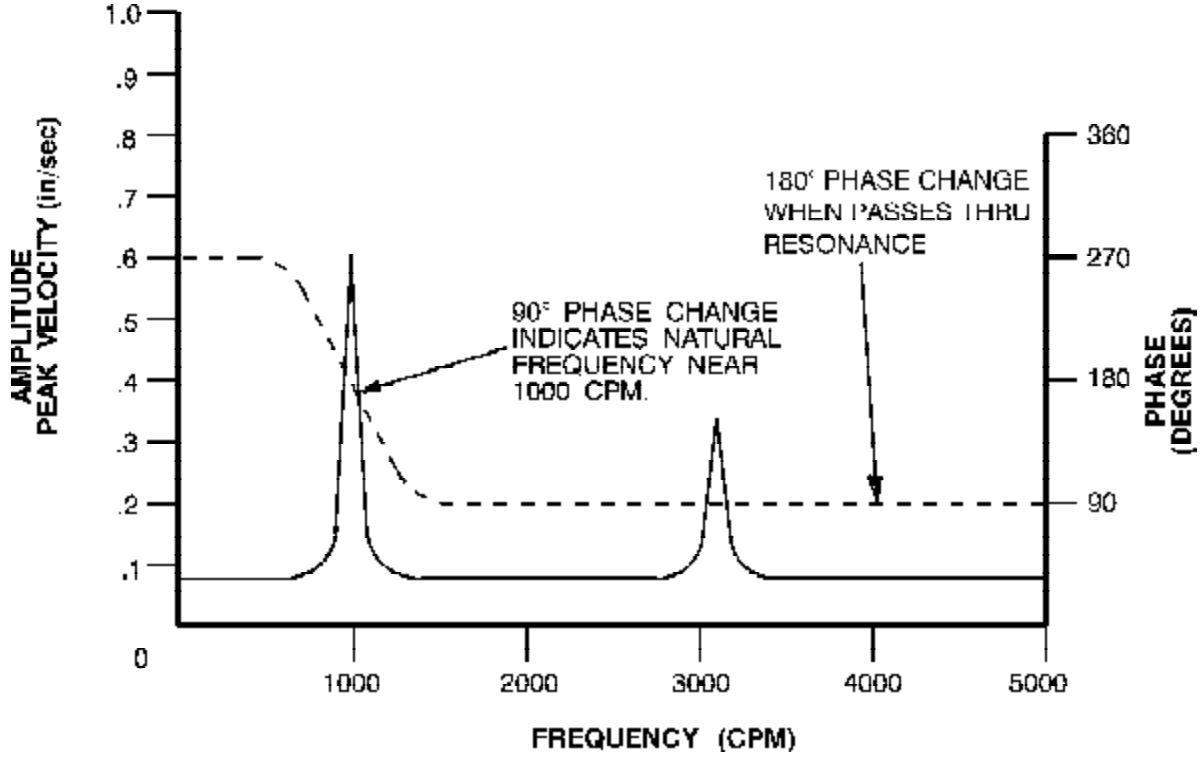


FIGURE 6.05C
BODE' PLOT OF SAME MACHINE IN FIGURE 6.05B

© Copyright 2001 Technical Associates Of Charlotte, P.C.

A runup test is identical to the coastdown except that the machine begins at rest and then is brought up to speed. Vibration and phase (Bode') measurements are taken during this runup and are plotted similar to the coastdown to determine the natural frequencies of the system.

3) OTHER METHODS- Other methods for determining natural frequencies include shaker tests,modal analysis, and finite element modeling. Shaker tests are conducted by attaching a variable speed motor with a preset unbalance to an object. The unbalance will force the object to which it is attached. Vibration response is monitored as the speed of the motor is "swept" through its range of frequencies. A high amplitude peak as well as a 90° phase change continuing on to 180° indicates a natural frequency. Modal analysis utilizes a force hammer (the input force can be recorded) a transducer, and a FFT device for recording the data. A computer graphical representation of the object being tested is created. The force input as well as the objects response to an impact from the hammer are recorded. The modal software computes the mode shapes and natural frequencies and displays them on the computer in an animated fashion. A finite element model consists of purely mathematical information to approximate the object's mode shapes and natural frequencies. Information such as the objects physical properties (mass, stiffness, etc...) and geometry are input into a finite element software program. The program then computes the estimated mode shapes and natural frequencies and displays them in an animated fashion on the computer. Note that the finite element method does not require any "field" information to be collected from the object, therefore, this method is very useful in estimating natural frequencies of objects in a design environment before the object is actually built.

Harmfulness

There is nothing harmful about a natural frequency unless a resonance occurs at a particular natural frequency.

6.052 Resonance

Simply speaking, resonance is a condition that occurs when a force is applied to an object at a particular frequency that is very near or equal to a natural frequency of the object. Also, the force or part of the force must be in the same direction as the natural frequency. The result of a resonance condition is a terrific increase in amplitude (10x - 30x higher) at the frequency that is resonant. The applied force may be from a number of sources. As was learned earlier in **Chapter 6**, unbalance creates a force at 1X RPM, misalignment creates a force at 1X, 2X, and 3X RPM, many other problems such as eccentricity or bent shaft create forces at 1X RPM, bearing problems create forces at noninteger harmonics of 1X RPM (i.e. - 4.64X RPM, 8.33X RPM, etc.), even motor electrical problems cause forces at frequencies such as 7200 CPM or at rotor bar pass frequency. If the frequency of any of these forces is equal to a machine's natural frequency and in the same direction, a resonance condition will be created. The result will be an increase in the vibration amplitude that would normally be generated by the problem on its own. It is important to remember 3 concepts:

1) A natural frequency that does not have a forcing frequency near it will not be excited (i.e. - no peak will be found in the frequency spectrum).

2) A problem with the machine will show up in the frequency spectrum and will have an amplitude that is based on the amount of force generated by the problem and the stiffness of the machine that is restraining the force provided that no natural frequency is present at that problem frequency.

3) If a problem occurs and its frequency is the same as one of the machine's natural frequencies, the amplitude in the frequency spectrum will be 10X to 30X higher than described in number 2 for the particular problem.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

6.06 MECHANICAL LOOSENESS

Many texts and seminars today simply refer to a general term called "Mechanical Looseness" and that it either can be detected by high vibration at 2X RPM or multiple running speed harmonics. However, a comprehensive study of a great amount of available literature as well as review of a number of case histories accomplished by Technical Associates actually has revealed that there are at least 3 different types of Mechanical Looseness, each of which has its own characteristic vibration spectra as well as vibration phase behavior. A discussion will follow on each of these which are as follows:

6.061 *Type A* - Structural Frame/Base Looseness (Primarily 1X RPM);

6.062 *Type B* - Looseness Due To Rocking Motion Or Cracked Structure/Bearing Pedestal (Primarily 2X RPM);

6.063 *Type C* - Loose Bearing In Housing or Improper Fit Between Component Parts (Multiple Harmonics Due To Nonlinearity Often Induced By Impulse Events)

One of the important facts about each type of mechanical looseness is that it alone is not a cause of vibration. Instead, looseness is a reaction to other problems which are present such as unbalance, misalignment, eccentricity, bearing problems, etc. Resolution of these other problems often will remove many of the symptoms, and therefore the response of looseness. However, the problem is that only minute amounts of such problems as misalignment or unbalance can cause vibration if a looseness condition exists. Looseness aggravates the situation. Therefore, mechanical looseness allows much more vibration than would otherwise occur from these other problems alone. Resolution of the other problems themselves will often remove most of the symptoms due to looseness. However, this is often virtually impossible in reality because such steps would require extraordinary levels of precision of alignment or balancing. Therefore, in these cases, the looseness condition will first have to be resolved. Then, if remaining vibration is still high, other steps such as alignment and balancing can be accomplished with much greater ease than before the looseness conditions were resolved. Following below is a discussion on each of the 3 types of looseness which were listed previously:

6.061 Type A - Structural Frame/Base Looseness (1X RPM):

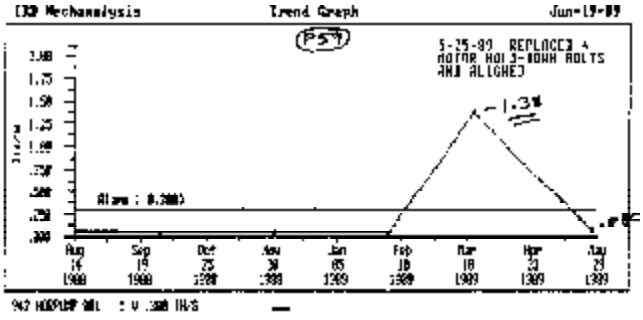
This type of looseness includes each of the following problems:

- Structural Looseness/Weakness of Machine Feet, Baseplate & Concrete Base;
- Deteriorated or Crumbled Grouting;
- Distortion of Frame or Base (Soft Foot);
- Loose Hold-Down Bolts.

Type A Looseness Problems are often misdiagnosed as unbalance or misalignment problems since they have almost identical vibration spectra. Therefore, it is important to look beyond vibration spectra and compare relative amplitudes between directions, look closely at phase behavior which departs radically from such things as unbalance, and to examine other characteristics which are listed as follows:

1. Type A Looseness spectra are dominated by high 1X RPM vibration and appear identical to that for an unbalance or eccentric rotor condition. A spectrum illustrating this type signature is shown in Figure 6.06A.

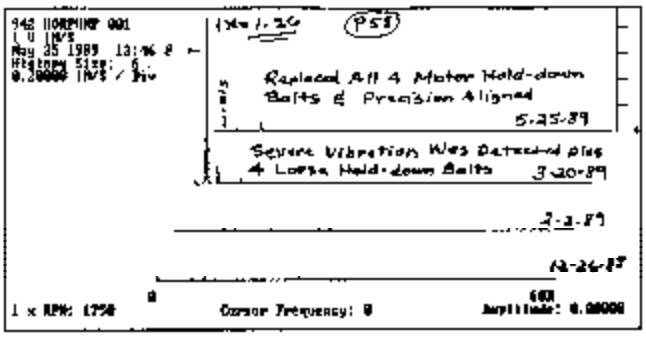
© Copyright 2001 Technical Associates Of Charlotte, P.C.



IRD Mechanalysis Data Analysis System

Machine 942 HORPUMP 001 Pos 1 Dir V Units IN/S Alarm .300

Ampl	Insp Code	Date mm/dd/yy	Time	Ampl	Insp Code	Date mm/dd/yy	Time
.062	0	08-14-88	10:01				
.054	0	08-15-88	13:00				
.042	0	09-29-88	11:03				
.038	0	12-01-88	9:49				
.041	0	02-02-89	15:00				
1.38	0	03-25-89	13:43				
.080	0	05-25-89	12:46				



001% x 2.0000 = 0.02000 1 x RPM: 1750.00

Machine	Pos	Dir	Units	Date
942 HORPUMP 001	1	V	IN/S	May 25 1989 12:46

Frequency	1 x RPM	Amplitude
3600	2.06	0.02478
7200	4.11	0.03782

3% x 2.0000 = 0.06000 1 x RPM: 1800.00

Machine	Pos	Dir	Units	Date
942 HORPUMP 001	1	V	IN/S	Mar 20 1989 13:43

Frequency	1 x RPM	Amplitude
1800	1.06	1.24665
3450	1.92	0.12994

001% x 2.0000 = 0.02000 1 x RPM: 1750.00

Machine	Pos	Dir	Units	Date
942 HORPUMP 001	1	V	IN/S	Feb 02 1989 13:00

Frequency	1 x RPM	Amplitude
21600	12.34	0.03940

1% x 2.0000 = 0.02000 1 x RPM: 1750.00

Machine	Pos	Dir	Units	Date
942 HORPUMP 001	1	V	IN/S	Dec 26 1988 10:01

Frequency	1 x RPM	Amplitude
150	0.09	0.03542
21600	12.34	0.02478

FIGURE 6.06A
SEVERE LOOSENESS INTRODUCED BY LOOSE HOLD-DOWN BOLTS

© Copyright 2001 Technical Associates Of Charlotte, P.C.

2. Often, high vibration is pretty well confined to only one rotor (that is, the driver or driven component or the gearbox alone). This is unlike unbalance or misalignment in which rather high vibration levels due to these other problems are not confined to just one of the rotors.

3. Two different phase behaviors can occur with Type A Looseness:

   a. When comparing vertical and horizontal phase on each of the bearing housings, the vibration will sometimes be found to be highly directional with phase differences of either 0° or 180° depending on whether or not the horizontal reading was taken on one side or the other (either a phase difference of 0° or 180° means that the motion is directly up and down or side to side). This does not normally occur with simple unbalance in which horizontal and vertical phase usually differs approximately 90° (±30°).

   b. When this first phase behavior occurs (0° or 180° phase difference in horizontal and vertical), the analyst should not confine his measurements to the bearing housings alone, but move on down to the machine foot, baseplate, concrete base and surrounding floor. This is illustrated in Figure 6.06B. Here, comparative amplitude and phase measurements should show relatively identical amplitude and phase at 1X RPM at each location. If there is a great difference in amplitude and phase, this will suggest relative motion. Using the point where this great phase change occurs, one can locate where the problem exists. For example, the measurements in Figure 6.06B show a problem between the baseplate and concrete base indicated by the great difference in phase (note 180° out of phase with the other two measurements). This indicates structural looseness/weakness allowing relative movement in machine components which may be due to a problem with the grouting between the baseplate and concrete base; or broken or cracked foundations, etc. On the other hand, if a great phase difference occurred between machine foot and baseplate, this might suggest looseness of the mounting bolt and/or possible stripped mounting bolt threads. Either of these two problem conditions can cause a great vibration at 1X RPM like that shown in Figure 6.06A on the machine component where this occurs, particularly on the bearing housing directly above this base location.

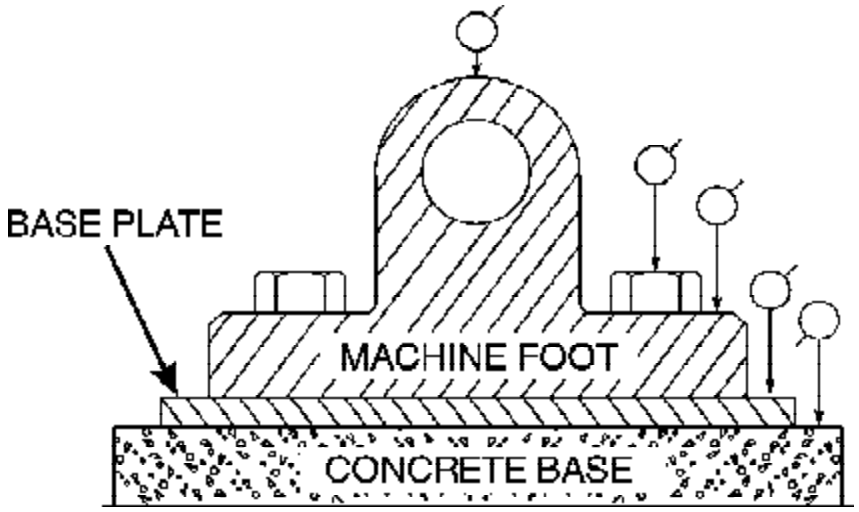


FIGURE 6.06B
PHASE ANALYSIS USED TO PINPOINT LOOSENESS

© Copyright 2001 Technical Associates Of Charlotte, P.C.

4. Figure 6.06C illustrates another case involving this high 1X RPM looseness vibration. In this case, bolts that are parallel to the pump shaft are mounted on each pump bearing housing acting in the axial direction. In this case, if these bolts become loose, it will cause a high vibration in the axial direction at 1X RPM which will closely resemble a misalignment problem. However, simply tightening these bolts will greatly reduce the vibration.

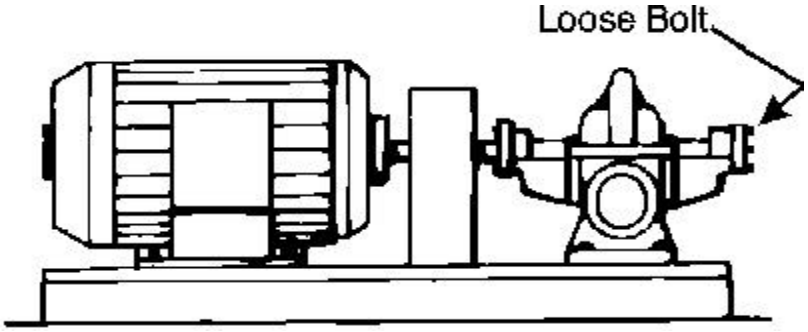


FIGURE 6.06C
EFFECT OF LOOSE BOLT ON PUMP OUTBOARD BEARING HOUSING

5. Distortion induced either by a soft foot or piping strain shows another situation which would result in high 1X RPM vibration signatures looking like unbalance. However, in these cases, when phase readings are taken, they will show highly directional vibration with the difference in horizontal and vertical phase approaching either 0° or 180° (±30°) rather than 90° in the case of simple unbalance. If the problem were distortion rather than looseness, amplitude and phase measurements would show the machine foot, baseplate and concrete base pretty much vibrating in the same direction (equal phase readings). However, they may show that the amplitude on either one of the foot bolts alone is much higher than that on any of the other 3 bolts; or, for example, the right front foot and left rear foot may be significantly higher than the left front and right rear feet. In this second case, the motor would be vibrating diagonally. Great amplitude differences on these foot bolts would suggest a soft foot which must be corrected to reduce the high vibration levels. (In fact, the analyst may find that if he backs off slightly on those bolts having high vibration, he may see much lower vibration than before).

6.062 Looseness Due To Rocking Motion or Cracked Structure/Bearing Pedestal (2X RPM):

The 2X RPM "looseness symptom" referenced in many vibration texts seems only to occur for the following looseness problems:

- o Crack in Structure or Bearing Pedestal;
- o Rocking Motion Sometimes Induced By Differential Length Support Legs;
- o Occasionally on Some Loose Bearing Housing Bolts;
- o When Loose Bearing or Improper Component Fit Problems Are Only of Minor Severity (No Impulse or Impact Events).

Figure 6.06D shows a spectrum typically showing these problems. These problems exhibit the following characteristics:

1. Typically, these problems are suggested when the amplitude at 2X RPM exceeds about 50% of that at 1X RPM in the radial direction.

2. Amplitudes are somewhat erratic.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

3. If phase readings are taken with a strobe light, it will often show 2 reference marks which are slightly erratic.

4. These looseness symptoms will not normally occur unless there is some other exciting force such as unbalance or misalignment. However, if this looseness condition exists, it will be extremely difficult to balance or align the unit sufficiently to bring down the final vibration sufficiently.

5. If the looseness problem is a bearing loose in the housing or a loose component on the shaft, the vibration will pretty well remain at 1X and 2X RPM until it worsens allowing an impulse or impact event. When this occurs, these impulses cause nonlinearities in the time waveform which will begin exciting many harmonics advancing to Type C Looseness.

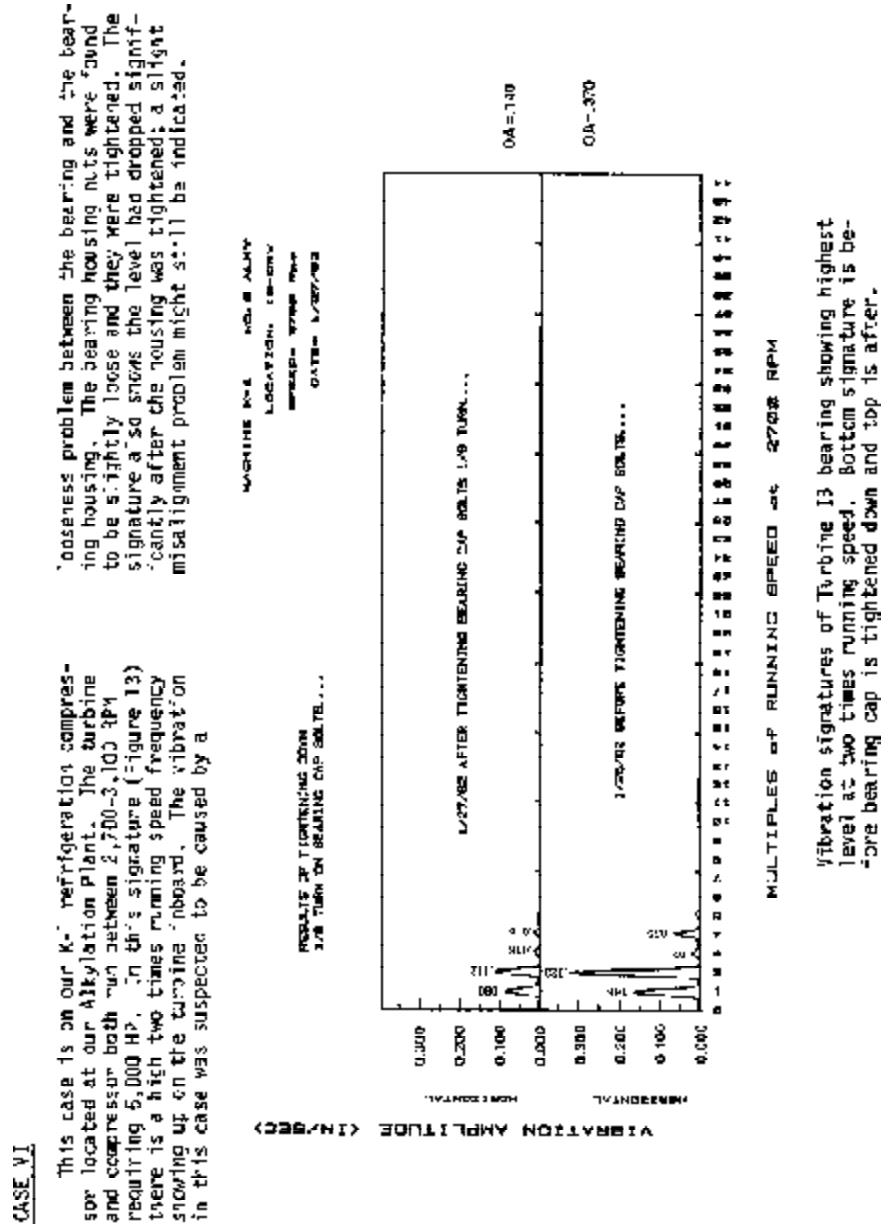


FIGURE 6.06D
EXAMPLE OF LOOSENESS OF FASTENERS OCCURRING AT 2X RPM
(Ref. 17)

© Copyright 2001 Technical Associates Of Charlotte, P.C.

6.063 Loose Bearing In Housing or Improper Fit Between Component Parts (Multiple Harmonics Due To Nonlinearity Often Induced By Impulse Events):

Each of the following problems occur in Type C Looseness:

- o Bearing Loose in Housing;
- o Excessive Internal Bearing Clearances;
- o Bearing Liner Loose In Its Cap;
- o Loose Rotor;
- o Bearing Loose and Turning on Shaft.

Figures 6.06E thru 6.06G illustrate typical spectra indicating Type C Looseness. Note the presence of multiple running speed harmonics in both spectra. Also note in Figure 6.06G that one of the running speed harmonics lies close to a natural frequency causing a resonant response at this frequency (if the looseness condition is resolved, the resonance condition will likely be as well). **Type C is the most common mechanical looseness problem** and exhibits the following characteristics:

1. Multiple running speed harmonics sometimes up to 10X or even 20X RPM are clearly present in this spectrum. These harmonics are a result of impulses and truncation (limiting) in the machine response. This impulse event causes a nonlinearity in the time waveform. When this occurs, multiple harmonics will appear in the resulting FFT spectrum (see Figure 6.06E).

2. This looseness tends to produce vibration that is directional which differs from unbalance. It normally will be highest in the direction and vicinity of the looseness problem. For example, this may show that the highest vibration is not either horizontal or vertical, but somewhere in between the two.

3. If the amplitude of harmonics becomes significant, this can also generate frequencies spaced at 1/2 times RPM (that is, .50X, 1.50X, 2.50X, etc) or even sometimes at 1/3 times RPM.

4. The analyst is cautioned that amplitudes of these 1/2 times RPM “harmonics” may appear deceptively low when compared with those at 1X RPM and running speed harmonics. However, he should remember that no peaks at 1/2 times RPM intervals should be present whatsoever. If the peaks are clearly evident, they do indicate a more advanced looseness problem (or possibly, presence of a rub).

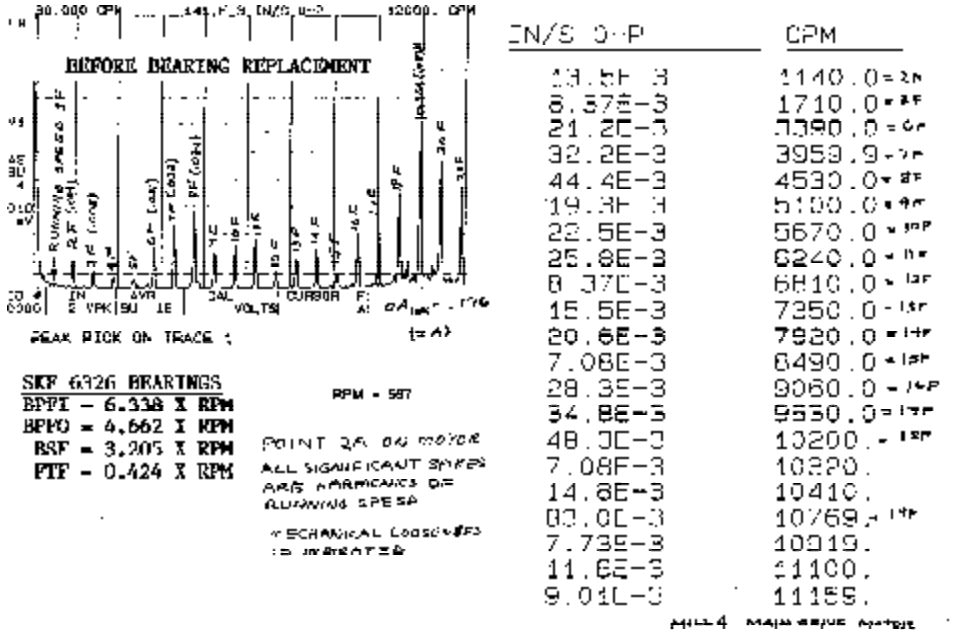
5. One half times RPM “harmonics” usually are accompanied by other problem sources such as unbalance and misalignment.

6. Phase measurements of Type C looseness problems are normally somewhat erratic, but can approach differences of 0° and 180° between horizontal and vertical directions if the vibration itself becomes highly directional. It normally acts in a radial direction, but can occur in the axial, dependent on the exact type of looseness.

7. In the case of a loose rotor such as a loose pump impeller, phase will vary from one startup to the next. The amplitude itself may be steady for a given run, but likewise will vary from startup to startup. Such a loose rotor is impossible to balance since the heavy spot itself is constantly changing directions. This shift in amplitude and phase is likely caused by center of gravity shifts.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

8. **CAUTION:** Vibration spectra simply appearing to be Type C looseness (many 1X RPM harmonics) can, in fact, signal a problem of much greater severity - a bearing loose and turning on a shaft. This can be the case even if amplitude of 1X RPM and its harmonics are fairly low, on the order of .05 in/sec or less. In these cases, the turning of the bearing on the shaft can actually cause great damage, actually removing material from the shaft diameter. In these cases, it can cause catastrophic failure of the machine as the bearing finally locks, and can do so without even generating any bearing defect frequencies. Figure 6.06E below is an example of just such a catastrophic failure where over .25 inch of the shaft diameter was removed from a 3.50 inch shaft before the bearing locked up.



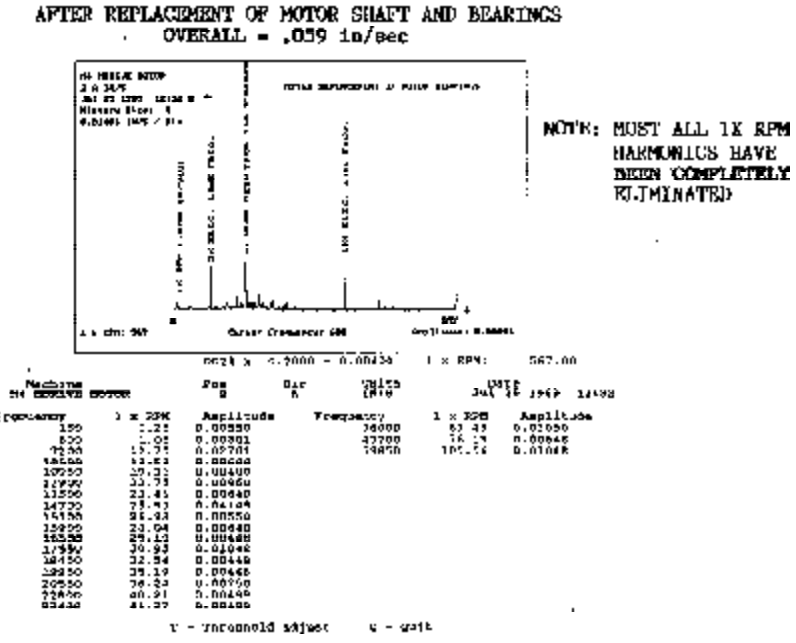
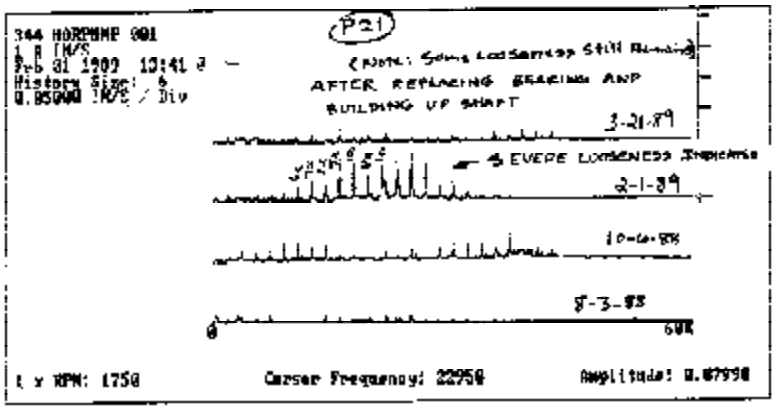


FIGURE 6.06E
BEFORE & AFTER REPLACEMENT OF A DC MOTOR BEARING WHICH WAS TURNING ON THE SHAFT RESULTING IN CATASTROPHIC FAILURE

© Copyright 2001 Technical Associates Of Charlotte, P.C.



22% x 0.5000 = 0.01000 1 x RPM: 1750.00

Machine	Pos	Dir	Units	Date
344 HORPUMP 001	1	A	IN/S	Mar 21 1989 12157

Frequency	1 x RPM	Amplitude
12300	7.03	0.01247
15900	9.09	0.01843
19500	11.14	0.01048
21600	12.34	0.01096
35250	20.14	0.01244
37050	21.17	0.01191
38850	22.20	0.01493
40650	23.23	0.01295
42450	24.26	0.01096

006% x 0.5000 = 0.03000 1 x RPM: 1750.00

Machine	Pos	Dir	Units	Date
344 HORPUMP 001	1	A	IN/S	Feb 01 1989 13141

Frequency	1 x RPM	Amplitude
12300	7.03	0.03002
14100	8.06	0.03503
15900	9.09	0.06497
17700	10.11	0.05997
19500	11.14	0.04249
21150	12.09	0.05496
22950	13.11	0.07998
24750	14.14	0.07498
26550	15.17	0.05496
28000	16.00	0.03002

2-1-89

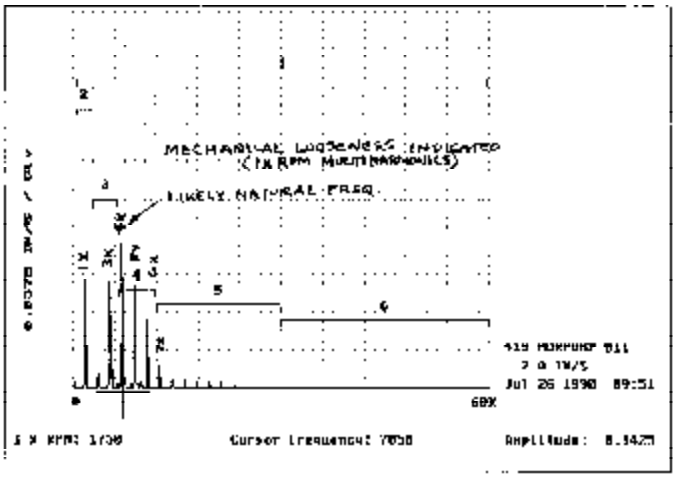
002% x 0.5000 = 0.01000 1 x RPM: 1750.00

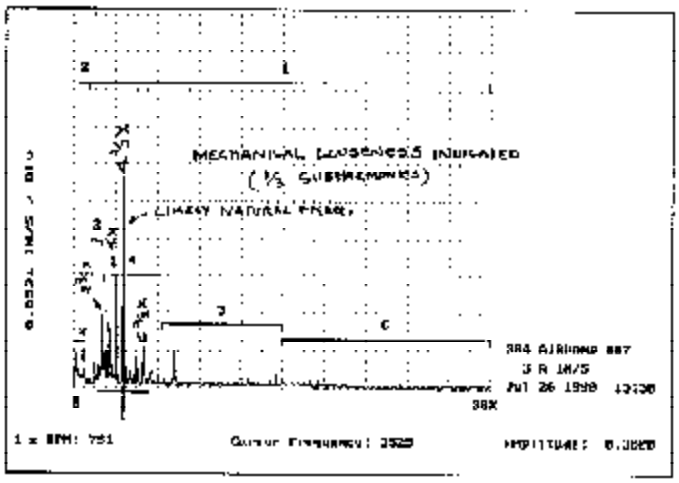
Machine	Pos	Dir	Units	Date
344 HORPUMP 001	1	A	IN/S	Aug 03 1988 07:27

Frequency	1 x RPM	Amplitude
600	0.17	0.01398
3000	1.71	0.01096
7200	4.11	0.01096
14100	8.06	0.01597
21450	12.26	0.01247

FIGURE 6.06F
ADVANCING PROBLEM WITH BEARING LOOSE ON SHAFT
(PARTIALLY RESOLVED)

© Copyright 2001 Technical Associates Of Charlotte, P.C.





0025 x 0.500 -- 0.0100 1 x RPM: 751.00

Machine	Pos	Dir	Units	Date
304 AIRHAND 007	3	A	IN/S	Jul 26 1990 15:50

Frequency	1 x RPM	Amplitude	Frequency	1 x RPM	Amplitude
150	0.20	0.0450	5550	7.39	0.0225= 7½X
750	1.00	0.0275= 1X	5700	7.59	0.0100
975	1.30	0.0100= 1½X	6000	7.99	0.0150= 8X
1125	1.50	0.0100= 1½X	6525	8.69	0.0125= 8½X
1500	2.00	0.0350= 2X	7200	9.59	0.0500= 9½X
1800	2.40	0.0300	12525	16.68	0.0125= 16½X
2025	2.70	0.1025= 2½X	14025	18.68	0.0100= 17½X
2250	3.00	0.0575= 3X	14550	19.37	0.0150= 18½X
2475	3.30	0.0900= 3½X	15000	19.97	0.0100= 19X
2775	3.70	0.0325= 3½X	17025	22.67	0.0100
3000	3.99	0.1550= 4X	17550	23.37	0.0100
3225	4.29	0.0150= 4½X			
3525	4.69	0.3000= 4½X			
4050	5.39	0.0225= 5½X			
4275	5.69	0.0150= 6½X			
4500	5.99	0.0425= 6X			
5025	6.69	0.0575= 6½X			

T - Threshold adjust Q - Quit

FIGURE 6.06G
2 DIFFERENT TYPES OF SPECTRA INDICATING
TYPE C MECHANICAL LOOSENESS

© Copyright 2001 Technical Associates Of Charlotte, P.C.

6.07 TRACKING OF ROLLING ELEMENT BEARING FAILURE STAGES USING VIBRATION AND HIGH FREQUENCY ENVELOPING AND DEMODULATED SPECTRAL TECHNIQUES

ABSTRACT:

Throughout the last decade, much research along with in-plant testing has been conducted in an attempt to accurately evaluate the condition of rolling element bearings. This has included studies to approximate remaining bearing life. This life is very dependent upon the vibration to which the bearing is subjected. The following paper will present a number of failure senores which have been identified to date for tracking rolling element bearing failure stages using both vibration and spike energy spectral analysis. These results have been directly correlated with studies which have been rigorously conducted in laboratories. Such studies have included evaluation of many types of rolling element bearings including deep groove ball, angular contact ball, needle, cylindrical roller, spherical roller and tapered roller bearings. It is the expressed purpose of this paper to provide the reader with solid tools with which he can not only evaluate the current health of specific rolling element bearings, but also can assist him in predicting remaining life and/or taking proactive steps immediately required to noticeably extend the life of the bearings.

INTRODUCTION:

A tremendous cross section of today's process and utility machinery is outfitted with rolling element bearings. In most all cases, these bearings are the most precise components within the machine, generally held to tolerances only 1/10th those of many of the remaining machine components. Yet, only about 10 to 20% of bearings achieve their design life due to a variety of factors. These primarily include lubrication inadequacies, use of the wrong lubricant, contamination with dirt and other foreign particles, improper storage outside their shipping packages, introduction of moisture, false brinelling during shipment or when standing idle, misapplication of the wrong bearing for the job, improper installation of bearings, etc.

Figure 6.07A shows the components of a rolling element bearing. Note the location of the accelerometer relative to the bearing's outer race, inner race, rolling elements and cage. When rolling element bearings wear, the vibration signal most readily travels from defects on the outer race to the accelerometer. Such flaws will normally appear on two or more of these components prior to eventual failure. Figure 6.07B illustrates various types of rolling element bearings.

One of the leading contributors to premature rolling element bearing failure is excessive vibration and the high dynamic loads that it can transmit into bearings. Following below is the design formula used in calculating theoretical ball bearing life which will show why it is so critical to bearing life to minimize the dynamic loads imposed upon them from vibration:

$$L_{10}\ \text{Life} = \left(\frac{16{,}666}{\text{RPM}}\right)\left(\frac{\text{RATING}_B}{\text{LOAD}_E}\right)^3 \quad \text{HOURS}$$

where:

L_{10} Life	=	No. of Hours that 90% of a group of bearings should attain or exceed prior to onset of fatigue failure.
RATING_B	=	Basic Dynamic Load Rating for a given bearing (lb)
LOAD_E	=	Equivalent Radial Load taken by a bearing - including Radial and Axial Loads (lb)

© Copyright 2001 Technical Associates Of Charlotte, P.C.

FIGURE 6.07A
ROLLING ELEMENT BEARING TERMINOLOGY

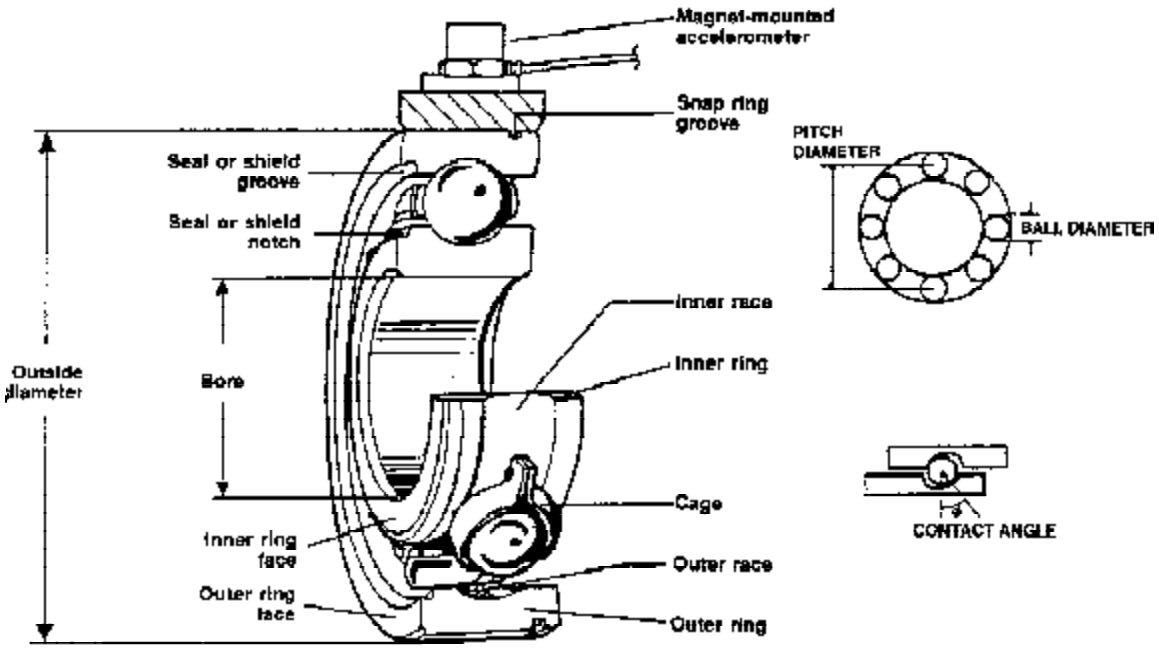
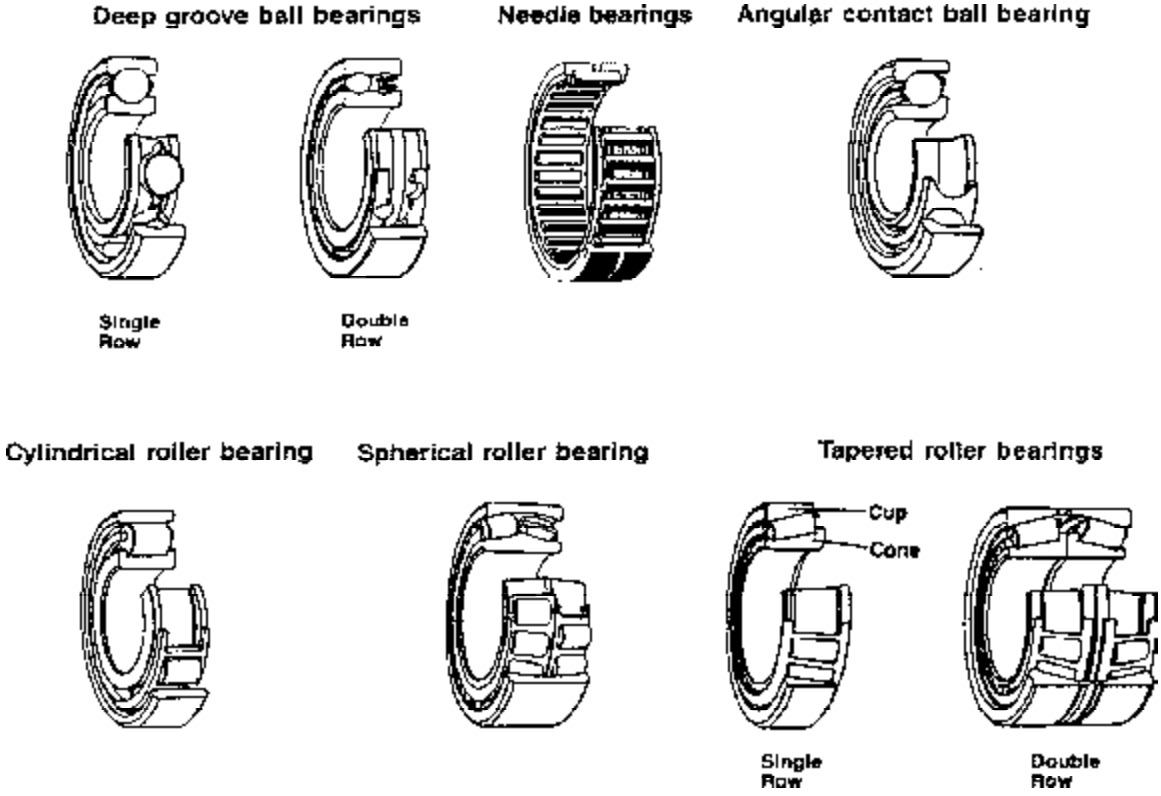


FIGURE 6.07B
ROLLING ELEMENT BEARING TYPES



© Copyright 2001 Technical Associates Of Charlotte, P.C.

This formula shows that the greater the speed, the lower the anticipated life. However, of even greater significance is that the theoretical ball bearing life varies with the 3rd power of the load to which it is subjected (and to the 3.33 power in the case of roller bearings). Therefore, if the designer only considered the static loads on the bearing as well as those from other components such as belt tension, he may be surprised to learn of the magnitude of the dynamic forces which can be introduced from vibration.

The key point is that rolling element bearings in truth rarely fail due to defective workmanship themselves. In most all cases, outside influences act on the bearing to bring about its premature failure. If outside influences such as unbalance, misalignment, belt drive problems, soft foot, inadequate lubrication and improper installation can be taken care of, the bearings themselves should have satisfactory life.

To bring home the point, if one considers only the load from unbalance, this alone can generate significant dynamic loads. For example, consider a 2000 lb rotor turning at 6000 RPM with 1 oz of unbalance on a 3 foot diameter (18" radius). The amount of centrifugal force from the unbalance alone can be calculated as follows:

$$F_C = mr\omega^2 = \left[\left(\frac{Wr}{(386)(16)}\right)\left(\frac{2\pi n}{60}\right)\right]^2$$

$$F_C = .000001775\ Un^2 = .00002841\ Wrn^2$$

where:

F_C = Centrifugal Force (lb)
U = Unbalance of Rotating Part (oz-in)
W = Weight of Rotating Part (lb)
r = eccentricity of the rotor (in)
n = Rotating Speed (RPM)

Now substituting for the sample rotor with a 1 oz unbalance at an 18" radius (U = 18 oz-in) turning 6000 RPM,

$$F_C = (.000001775)(18\ \text{oz-in})(6000\ \text{RPM})^2$$

F_C = 1150 lbs (from centrifugal force due to unbalance alone)

That is, only a 1 oz unbalance on a 3 foot diameter wheel turning 6000 RPM would introduce a centrifugal force of 1150 lbs which would have to be supported by the bearings in addition to the 2000 lb static rotor weight. Therefore, if the designer had only anticipated supporting 2000 lbs by the bearings, but in fact had to withstand 3150 lbs, his design life calculation would be off by a factor of:

$$\text{Corrected } L_{10} \text{ Life} = (\text{Initial Life})\left(\frac{2000}{3150}\right)^3$$

Corrected L_{10} Life = .25 X Initial Life Calculation (only 25% of design life)

Therefore, the actual theoretical life would only be 25% of his initial design life if this wheel were subjected only to unbalance, not to mention other dynamic forces introduced from vibration due to misalignment, looseness, cavitation or any other problems.

Of great importance is the ability to "track" the condition of rolling element bearings and to know when they will need replacement, right from the beginning when initial baseline signatures are

© Copyright 2001 Technical Associates Of Charlotte, P.C.

acquired. Much has been learned during the past decade on how vibration signature analysis can contribute to this capability. Following in Section 6.073 will be a separate illustrated discussion on how this can be accomplished using not only vibration signature analysis, but also some of the newer high frequency enveloping signal processing techniques as well. This discussion will follow sections on "Which vibration parameter should normally be used?" (Section 6.071) and "What types of vibration frequencies are generated by defective rolling element bearings?" (Section 6.072).

6.071 Optimum Vibration Parameter For Rolling Element Bearing Condition Evaluation (Acceleration, Velocity Or Displacement)?

Particularly during this last decade, we have learned that we certainly can no longer depend on overall vibration alone to accurately evaluate the health of rolling element bearings. Also, we have found we cannot depend only on the measurement of ultrasonic frequency broadband measurements (Spike Energy, HFD, Shock Pulse, etc.) which make measurements in approximately the 5000 Hz to 60,000 Hz region. What we have learned is that these ultrasonic measurements are only an indicator, not the indicator of bearing health. They likewise are most effectively used in conjunction with vibration signature analysis to best evaluate bearing condition.

Of course, when employing vibration signature analysis, either of 3 vibration parameters can be used - acceleration (g), velocity (in/sec) or displacement (mils). Figure 6.07C shows how each of these parameters vary with frequency in terms of severity. Following below are comments on the attributes of each of these vibration parameters when specifically evaluating rolling element bearing health:

A. Displacement - Unfortunately, displacement spectra miss a great deal of bearing health information. Since displacement is low frequency intensive, it tends to suppress or almost eliminate much of the spectral content available that indicates bearing defect problems. In fact, one of the real problems with employing displacement on low-speed machines less than 200 RPM is the fact that while the spectrum may successfully display 1X RPM, it most often will almost completely miss bearing frequencies until the problem is quite severe.

B. Acceleration - Unlike displacement, acceleration tends to overemphasize much of the high frequency content generated by the rolling element bearing defects. As a result, if one is not greatly familiar with working with it, acceleration spectra might "cry wolf" far too often. For plant programs trying to establish themselves, this can do great harm to credibility. Although acceleration itself is probably a better indicator in the very early stages of bearing problems, it quickly gives way to vibration velocity which more accurately and clearly tells the "true story" of current bearing health.

C. Velocity - Velocity spectra should be one of the best parameters for evaluating most bearing problems, even on low-speed machines (for example, even if the speed was only about 60 RPM, much of the bearing frequency content would be above 500 CPM). For common rotational speeds ranging from 1200 to 3600 RPM, most of the spectral vibration content containing bearing defect information will be below 2000 Hz (120,000 CPM). In general, depending on the type of transducer employed, velocity will remain somewhat "flat" in frequencies ranging from 600 CPM up to 120,000 CPM (10 Hz - 2000 Hz). This means that somewhat "equal weight" can be assessed to a bearing defect frequency occurring at either 6000 CPM or at 60,000 CPM, whereas this could not be done with acceleration or displacement which are highly frequency dependent (see Figure 6.07C). When rolling element bearing machines operate at speeds above approximately 10,000 RPM, acceleration would then likely be the best rolling element bearing health indicator.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

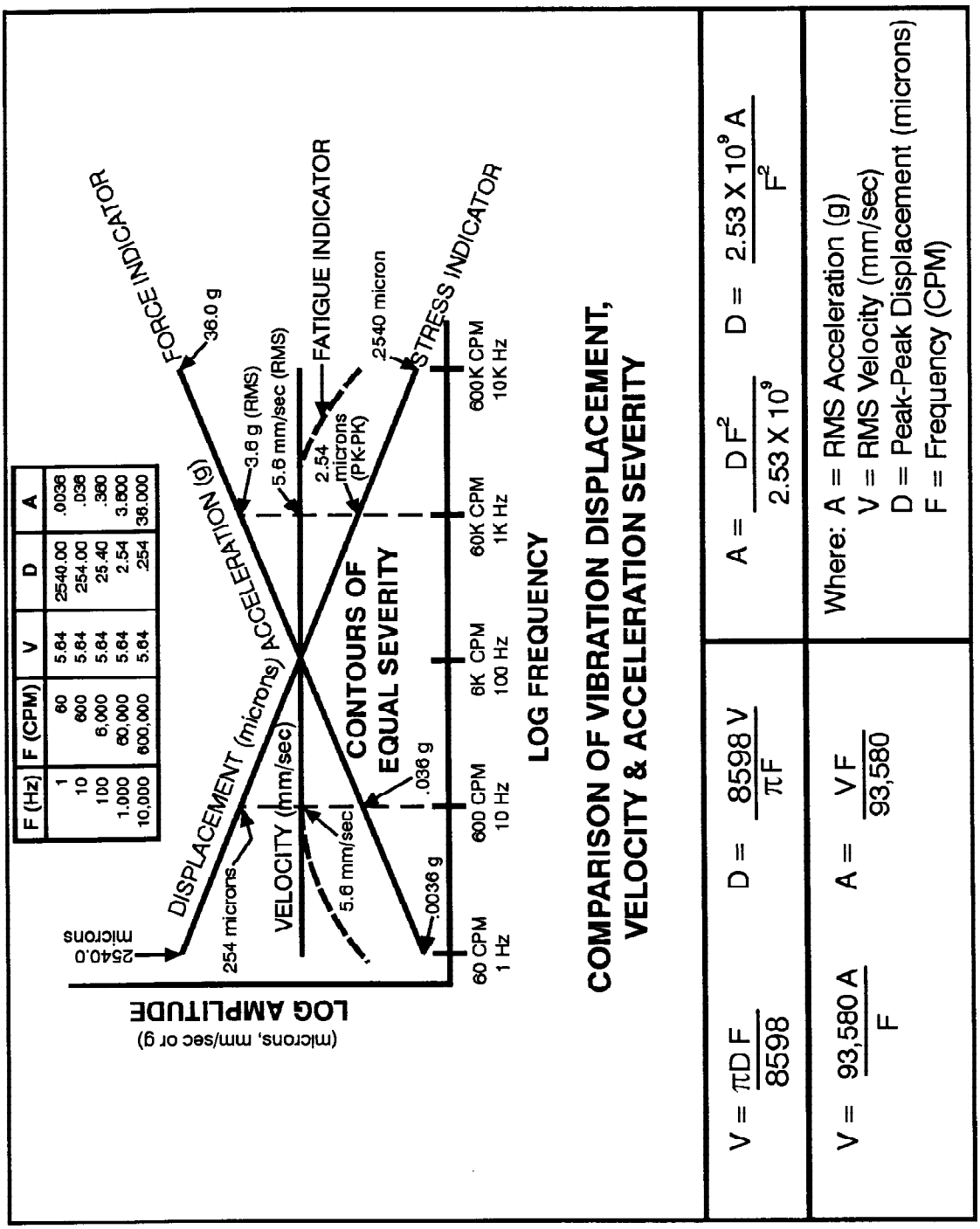


FIGURE 6.07C

© Copyright 2001 Technical Associates Of Charlotte, P.C.

The remainder of this section on rolling element bearing health will assume velocity spectra. However, if one takes into account how the other parameters such as acceleration vary with frequency, he can apply many of these same techniques. Acceleration spectra may detect such bearing problems earlier than will velocity spectra, particularly on high-speed machinery. In addition, high frequency envelope demodulated spectra will likely provide even earlier warning of bearing wear and lubrication problems, and should be employed in addition to vibration spectra alone on critical machinery (this will be discussed in Section 6.073 of the paper).

6.072 Types Of Vibration Spectra Caused By Defective Rolling Element Bearings:

Defective rolling element bearings generate each of 4 types of frequencies once they begin to develop defects. These frequencies include (a) random, ultrasonic frequencies; (b) natural frequencies of bearing components; (c) rotational defect frequencies; and (d) sum and difference frequencies. Following below will be a discussion on each of these types of frequencies and their significance:

(a) Random, Ultrasonic Frequencies:

Measurements in this ultrasonic frequency region ranging from approximately 5000 Hz to 60,000 Hz are made by a variety of instruments and employ a similar variety of techniques. These include spike energy, HFD high frequency acceleration, shock pulse measurement and others. Each of these are meant to be incipient failure detection parameters that can track bearing health from its installation until just prior to or including eventual failure. Each of them have their own strengths and weaknesses, not only concerning their bearing health evaluation accuracy, but also how well each of them can be trended with time. In general, the overall number they provide gives just one more piece of information to be considered when evaluating bearing health. However, the information contributed by the vibration spectral data should be given significantly more weight.

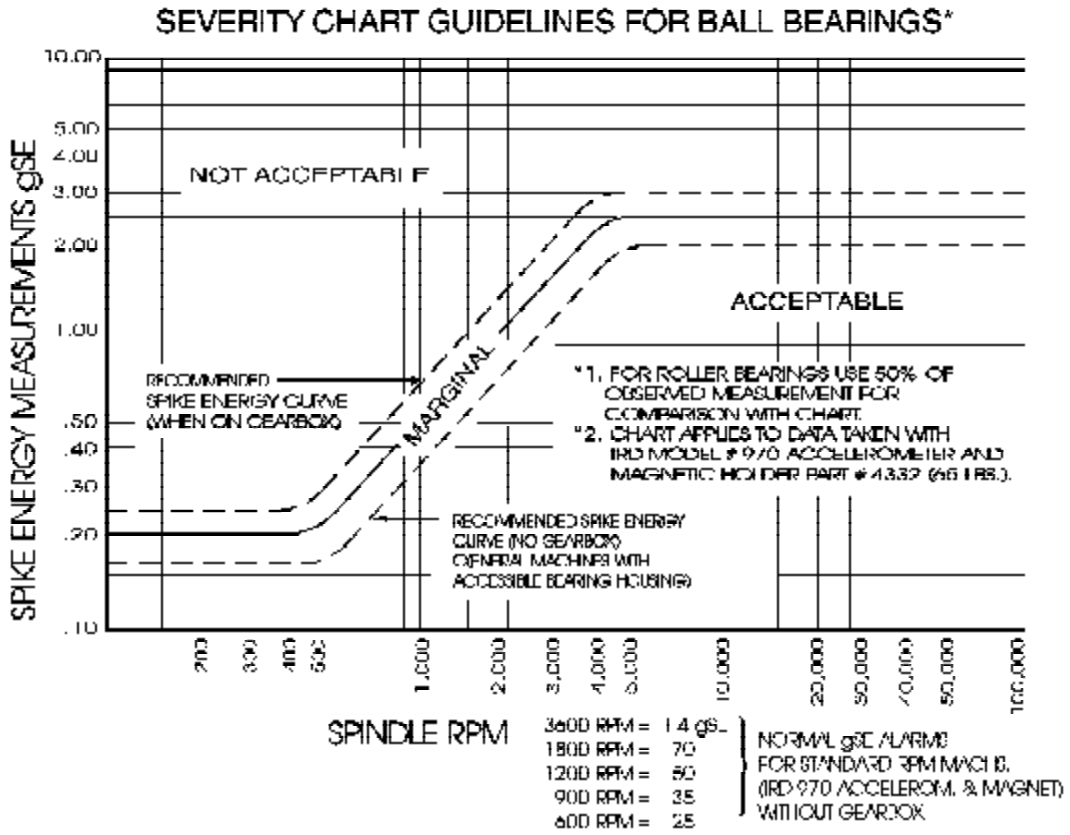
Figure 6.07D provides a severity chart for specifying alarm levels of Spike Energy. This chart shows that machine speed must be taken into account when evaluating Spike Energy amplitudes. Similarly, Figure 6.07E provides a comparable severity chart for “HFD” and Shock Pulse (SPM), comparing the relative amplitudes of these parameters with Spike Energy levels in a study conducted by Mr. Charles Berggren (References 16 & 19).

(b) Natural Frequencies of Bearing Components (when installed):

Reference 11 documents that the natural frequencies of installed rolling element bearing components range from approximately 500 to 2000 Hz (30,000 to 120,000 CPM). Like every other member, these bearing components “ring” at these natural frequencies when they are impacted. In the case of rolling element bearings, intermittent impacts of the rolling elements striking flaws on the raceways ring their natural frequencies. Actually, there are several bearing component natural frequencies in the region of 30,000 to 120,000 CPM, but some are much more predominant than others. Therefore, when defects progress beyond microscopic size, they begin to excite these natural frequencies making them the “second line of detection” (Failure Stage 2, discussed in Section 6.073). As the defects worsen, they can cause greater impacts which cause greater response from the natural frequency peaks. Eventually, when wear progresses, more frequencies around these resonances appear, many of which will be 1X RPM sidebands of these natural frequencies (often, such modulating peaks will be spaced at bearing defect frequencies rather than 1X RPM sidebands).

© Copyright 2001 Technical Associates Of Charlotte, P.C.

FIGURE 6.07D
RECOMMENDED SPIKE ENERGY SEVERITY CHART (IRD SPIKE ENERGY)
(Ref. 25)



SPIKE ENERGY™ MEASUREMENTS

Energy is generated by repetitive transient mechanical impacts. Such impacts typically occur as a result of surface flaws in rolling-element bearings or gear teeth.

This energy is conducted from its source through various paths to the outer surface of the machine structure, and is seen as a small-amplitude vibration at the surface. Accelerometers coupled to the surface generate corresponding electrical signal.

The accelerometer signals processed by unique filtering and detection circuitry to produce a single "figure of merit" related to the intensity of the original impacts. This figure of merit is expressed in "gSE" units.

SPIKE ENERGY™ gSE readings are measurements which can with experience, be correlated with the severity of the casual surface flaws. Even though gSE readings are affected by the nature of the conductive path between the impact source and the accelerometer, similar machine structures will provide a reasonable basis for comparison between the structures.

The gSE figure of merit has proven to be effective in detecting mechanical defects in meshing gears and rolling element bearings. The gSE measurement, when used in conjunction with conventional measurement of vibration velocity and acceleration, provides early indications of mechanical deterioration.

** When used with magnetic holders, accelerometers must be installed with a light coating of silicone grease and tightened to 40 in-lb. torque.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

FIGURE 6.07E
MAINTENANCE DIAGNOSTIC VIBRATION AND HIGH FREQUENCY GENERAL TOLERANCE CHART FOR PROCESS MACHINERY WITH ROLLING ELEMENT BEARINGS①

Unfiltered Vibration Velocity	Shock Pulse & Acoustic Emission	Microlog HFD	IRD Spike Energy②	Severity Level and Alarm Status
Inches/Second Peak	Db	g	gSE	
1.5+	50+	5.0+	3.0+	Danger (Shutdown)
.75 - 1.49	40 - 49	3.0 - 4.99	1.50 - 2.99	Very Rough (Alert)
.40 - .74	30 - 39	1.50 - 2.99	.80 - 1.49	Rough (Alert)
.20 - .39	20 - 29	.75 - 1.49	.40 - .79	Fair (Acceptable)
.10 - .19	10 - 19	.30 - .74	.20 - .39	Good (Acceptable)
.01 - .09	1 - 9	.01 - .29	.01 - .19	Smooth (Acceptable)

① Provide by Charles Berggren of Monsanto
② Spike Energy Amplitudes measured using an IRD 970 accelerometer outfitted with IRD's 2-pole, 65 lb. magnet.

One important point about these bearing component natural frequencies is the fact that they are independent of running speed. That is, whether the shaft is turning at low or at very high speeds, the natural frequencies remain at the same frequency location. However, their response amplitude is proportional to the impact velocity, which means that with greater rotational speeds, they normally will respond at greater amplitudes.

(c) Rotational Defect Frequencies:

Through the years, a series of formulas have been developed which can help detect specific defects within rolling element bearings. They can separately detect faults on the inner race, outer race, cage or rolling elements themselves. They are based on the bearing geometry, the number of rolling elements and the bearing rotational speed. Figure 6.07F provides the formulas for each of these four rolling element bearing defect frequencies.

The power of these equations is that if one knows the design parameters of his bearings (pitch diameter, rolling element diameter, number of rolling elements & contact angle), he is able to detect problems which occur on the races, cage or rolling elements, and he is enabled to track these problems as deterioration continues. In many cases, the analyst may not know all the parameters to insert for a particular bearing in the equations, but he might know the bearing manufacturer and model number. In these cases, there are several publications and software offerings which tabulate each of the 4 defect frequencies for each of the bearing model numbers. Figure 6.07G is an example of one of the better known publications listing these defect frequencies (as per Reference 12). Note on Figure 6.07G that this sheet provides the number of rolling elements (N_b), rolling element diameter (B_d), bearing pitch diameter (P_d), contact angle

© Copyright 2001 Technical Associates Of Charlotte, P.C.

(∅), outer race defect frequency (BPFO), inner race defect frequency (BPFI), cage defect frequency (FTF) and rolling element defect frequency (BSF). Note that Figure 6.07G also provides an example assuming the SKF N220 bearing. Note that each one of the bearing frequencies are given in terms of running speed orders. Therefore, the frequencies for the SKF N220 bearing show, for example, that the inner race frequency (BPFI) is 9.13X RPM. Thus, if this bearing were turning at 1000 RPM, and a frequency was detected at 9130 CPM, he could conclude that this peak is the bearing inner race defect frequency and that a fault is developing there. On the other hand, if a frequency of 6860 CPM occurred on this same unit, he would know this is the outer race defect frequency of this SKF N220 bearing (since BPFO = 6.86X RPM).

The great advantage of knowing these bearing defect frequencies is that, for example, an analyst can separately evaluate the outboard and inboard bearings on the same machine, particularly if they are different model numbers. For example, assume that the outboard bearing of our example was an SKF N220 and that its inboard bearing was an SKF N228 bearing. Note from Figure 6.07G that they would have an entirely different set of defect frequencies (for example, the inner race frequency on the N228 would be 10.19X RPM as compared to 9.13X RPM on the N220).

A number of interesting facts can be stated about these bearing defect frequencies as follows:

1. How Bearing Frequencies Differ From Other Frequencies (Defect Frequencies):

One thing setting rolling element bearing defect frequencies apart from other vibration sources is the fact that they are defect frequencies. In other words, bearing defect frequencies should not be present. When they are present, they signal at least an incipient problem. On the other hand, other common frequencies such as 1X RPM are always present whether or not there is satisfactory or unsatisfactory balance or alignment; pumps and vanes always show vibration at some amplitude for blade pass frequencies; gears cause vibration at the number of teeth X RPM. However, the presence of these other frequencies does not mean there is necessarily a defect or problem. The appearance of bearing defect frequencies sends a message to the analyst to "pay attention". However, it is also important to point out the presence of such defect frequencies do not necessarily mean there are defects within the bearing. They also will appear if there is insufficient lubricant allowing metal-to-metal contact, or if the bearings are improperly loaded (excessive press fit, excessive thrust on a bearing not necessarily designed to take thrust, or if a thrust bearing is installed backwards, etc.).

2. Bearing Defect Frequencies are Noninteger Multiples of Operating Speed:

Referring back to Figure 6.07G, note that each of the bearing defect frequencies are noninteger multiples. That is, they are one of the few machinery vibration sources that do not generate integer multiples of rotational speed (GMF = #teeth X RPM; BPF = #blades X RPM). This is helpful when the manufacturer and model number of the bearing is unknown. Look for real number RPM multiples (such as 5.78X, or 7.14X, etc.).

3. Sum of Race Frequencies = "Bearing Ball Pass Frequency":

Figure 6.07F shows an interesting relationship between the outer and inner race frequency multipliers and the number of rolling elements. Note that the product of the number of balls X RPM equals the sum of the outer and inner race frequencies (BPFO + BPFI). For years, many have looked for the appearance of frequencies at the number of balls X RPM similar to what they have experienced with gears (#teeth X RPM), blade pass frequency (#Blades X RPM) and so forth. However, the author has rarely seen the appearance of a frequency at "bearing ball pass frequency", but has commonly seen each of the race frequencies themselves which are on the other side of the equation.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

FIGURE 6.07F
ROLLING ELEMENT BEARING DEFECT FREQUENCIES

These equations assume the inner race is rotating with the shaft while the outer race is fixed (stationary):

INNER RACE = BPFI = $\frac{N_b}{2}\left(1+\frac{B_d}{P_d}COS\ \theta\right) X\ RPM$

OUTER RACE = BPFO = $\frac{N_b}{2}\left(1-\frac{B_d}{P_d}COS\ \theta\right) X\ RPM = N_b\ X\ FTF$

BALL (OR ROLLER) = BSF = $\frac{P_d}{2B_d}\left[1-\left(\frac{B_d}{P_d}COS\ \theta\right)^2\right] X\ RPM$

CAGE = FTF = $\frac{1}{2}\left(1-\frac{B_d}{P_d}COS\ \theta\right) X\ RPM = .55 - .45X\ RPM$

NOTE:
If the inner ring is fixed (stationary) while the outer race rotates with the shaft, the minus sign must be changed to a plus sign within the parentheses of the Cage Frequency Equation. In this case, N_b X FTF will now equal BPFI rather than BPFO; and FTF = .55 + .65X RPM. However, BPFO, BPFI and BSF will remain the same.

(NOTE THAT BPFI − BPFO = N_b)

where:

N_b − Number of Balls or Rollers

B_d = Ball or Roller Diameter (in. or mm)

P_d = Bearing Pitch Diameter (in. or mm)

θ = Contact Angle (degrees)

APPROXIMATIONS (If Bearing Dimensions are Unknown, but do know # of Balls or Rollers) (Reference 18):

Approx. BPFI ≈ $\left(\frac{N_b}{2}+1.2\right) X\ RPM$

Approx. BPFO ≈ $\left(\frac{N_b}{2}-1.2\right) X\ RPM$

Approx. BSF ≈ $\frac{1}{2}\left(\frac{N_b}{2}-\frac{1.2}{N_b}\right) X\ RPM$

Approx. FTF ≈ $\left(\frac{1}{2}-\frac{1.2}{N_b}\right) X\ RPM$

© Copyright 2001 Technical Associates Of Charlotte, P.C.

FIGURE 6.07G
EXAMPLE TABULATION OF ROLLING ELEMENT BEARING DEFECT FREQUENCIES (REFERENCE 12)

SKF

SINGLE ROW CYLINDRICAL ROLLER BEARINGS

BEARING NUMBER	No	Bd	Pd	α	BPFO	BPFI	FTF	BSF
N205	12	.256	1.516	0	4.99	7.01	.41	2.67
N206	12	.295	1.811	0	5.00	6.94	.41	2.99
N207	12	.354	2.106	0	4.97	6.98	.41	2.89
N208	13	.394	2.362	0	5.42	7.58	.41	2.92
N209	14	.394	2.559	0	5.97	8.07	.42	3.17
N210	15	.394	2.968	0	6.50	8.48	.43	3.70
N211	15	.433	3.051	0	6.43	8.56	.43	3.45
N212	16	.472	3.476	0	6.92	9.08	.43	3.61
N213	16	.512	3.641	0	6.88	9.11	.43	3.49
N214	16	.512	3.738	0	6.93	9.06	.43	3.68
N215	17	.551	4.035	0	7.34	9.69	.43	3.59
N216	16	.590	4.331	0	6.91	9.08	.43	3.60
N217	16	.620	4.625	0	6.91	9.08	.43	3.60
N218	16	.709	4.921	0	6.84	9.15	.43	3.40
N219	16	.748	5.215	0	6.85	9.14	.43	3.41
N220	15	.787	5.512	0	6.86	9.13	.43	3.43
N222	15	.905	6.102	0	6.81	9.18	.42	3.59
N224	15	.945	6.594	0	6.85	9.14	.42	3.41
N225	18	.945	7.087	0	7.91	10.07	.43	3.32
N228	18	1.024	7.677	0	7.76	10.19	.43	3.68
M305	10	.354	1.713	0	4.01	5.99	.40	2.32
M306	11	.394	2.007	0	4.42	6.58	.40	2.45
M307	11	.433	2.264	0	4.45	6.55	.40	2.51
M308	12	.472	2.559	0	4.88	7.07	.41	2.62
M309	11	.551	2.854	0	4.43	6.56	.40	2.49
M310	12	.590	3.150	0	4.85	7.09	.40	2.57
M311	11	.669	3.445	0	4.42	6.56	.40	2.48
M312	12	.709	3.740	0	4.85	7.10	.41	2.54
M313	12	.748	4.035	0	4.87	7.08	.41	2.60
M314	12	.787	4.330	0	4.89	7.06	.41	2.66
M315	12	.866	4.626	0	4.85	7.09	.41	2.57
M316	13	.866	4.921	0	5.36	7.65	.41	2.75
M317	12	.945	5.216	0	4.89	7.06	.41	2.67
M318	13	.984	5.512	0	5.34	7.65	.41	2.71
M319	14	1.024	5.807	0	5.77	8.22	.41	2.74
M320	14	1.102	6.201	0	5.75	8.24	.41	2.68
M321	14	1.181	6.496	0	5.72	8.27	.41	2.66
M322	14	1.260	6.890	0	5.72	8.27	.41	2.64

EXAMPLE:
FOR SKF N220 BEARING, BPFI = 9.13 X RPM (Inner Race)
BPFO = 6.86 X RPM (Outer Race)
BSF = 3.43 X RPM (Rolling Elements)
FTF = .43 X RPM (Cage Freq.)

© Copyright 2001 Technical Associates Of Charlotte, P.C.

4. Description of How Defect Frequencies are Generated Within Rolling Element Bearings:

Figure 6.07F provides the formulas required to calculate the four defect frequencies for each rolling element bearing (BPFI, BPFO, BSF and FTF). Figure 6.07H illustrates how such defect frequencies are generated within the bearings. For example, if there is a defect on the outer race on the bottom of the bearing within the load zone as shown in Figure 6.07H, note that an impulse occurs in the time waveform at each instant when a rolling element passes over and impacts this defect. On the other hand, if the inner race had a defect, an impulse would occur in the time domain when the inner race rotated past each rolling element (assuming the inner race is press fit on the shaft). An important fact shown by Figure 6.07H is that the amount of response from the rolling elements striking the inner race defect will depend on where the inner race is positioned at that particular instant of time when the impact occurs (that is, if the inner race defect is positioned within the load zone, it will have significantly more response than it would if the impact occurred with the same inner race defect positioned 180° away, clearly out of the load zone). This explains why inner race defect frequencies are often surrounded by sidebands spaced at 1X RPM since their amplitude is modulated at the rate of once per revolution as shown by Figure 6.07H. On the other hand, since the outer race does not rotate in this instance, the amplitude response in the time domain should remain near constant. Therefore, 1X RPM sidebands surrounding outer race frequencies (BPFO) are much more serious than those surrounding the inner race frequency (BPFI), again assuming the inner race is press fit on the shaft (if the inner race is stationary while the outer race rotates, the reverse with respect to sidebands would occur). The presence of 1X RPM sidebands surrounding outer race (BPFO) frequencies normally means the problem is sufficiently serious to cause the bearing to actually impeed the motion of the shaft. As Section 6.073 will later point out, when the inner race frequency (BPFI) becomes surrounded by several "families" of 1X RPM sidebands, this can likewise indicate a more serious problem.

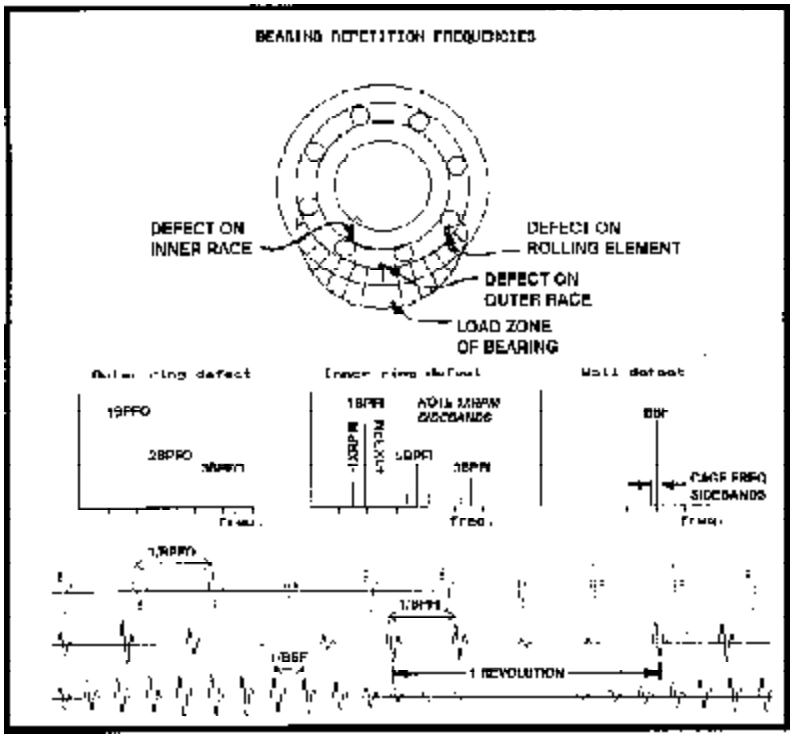


FIGURE 6.07H
ILLUSTRATION OF HOW DEFECT FREQUENCIES ARE GENERATED WITH ROLLING ELEMENT BEARINGS

© Copyright 2001 Technical Associates Of Charlotte, P.C.

5. Relative Outer and Inner Race Amplitudes:

Normally, the amplitude of the outer race frequency is higher than that of the inner race frequency. This is likely due to the fact that the transducer itself is much closer to the outer race (see Figure 6.07A). In addition, the vibration signal from the inner race frequency (or harmonics) must pass through several more interfaces including the constantly rotating rolling elements on its path out to the vibration transducer.

6. Usual Order of Appearance by Defect Frequencies:

Normally, defects will first appear on the races before the problem propagates to the rolling elements and cage. Thus, outer and inner race frequencies are normally first established before the appearance of a ball spin frequency. Later, the cage frequency normally can appear either as a fundamental or as a sideband to another frequency. Likewise, the ball spin frequency will sometimes appear as a sideband above and below an outer or inner race defect frequency.

7. Where Cage Frequency Normally Appears:

Although rolling element bearing problems are classically thought of as high frequency problems, fundamental cage frequencies will always be subsynchronous, ranging from approximately .33X RPM up to .48X RPM with the majority falling between .35X and .45X RPM. However, the cage frequency will not normally appear at its fundamental frequency. Instead, it most often will appear as a sideband around ball spin frequency (BSF), or around one of the race frequencies (BPFO or BPFI) with the sideband difference frequency equal to the cage frequency (FTF). For example, see Figure 6.09I which shows an outer race frequency BPFO) at 10,260 CPM (5.78X RPM) with cage frequency sidebands (FTF) spaced at 720 CPM (.41 X running speed which is 1775 RPM). This is how a cage frequency most often will appear.

8. Frequencies Generated by Faults on Balls or Rollers:

When defects occur on rolling elements themselves, they will often generate a frequency not only at the ball spin frequency (BSF), but also at the cage frequency (FTF, also known as fundamental train frequency).

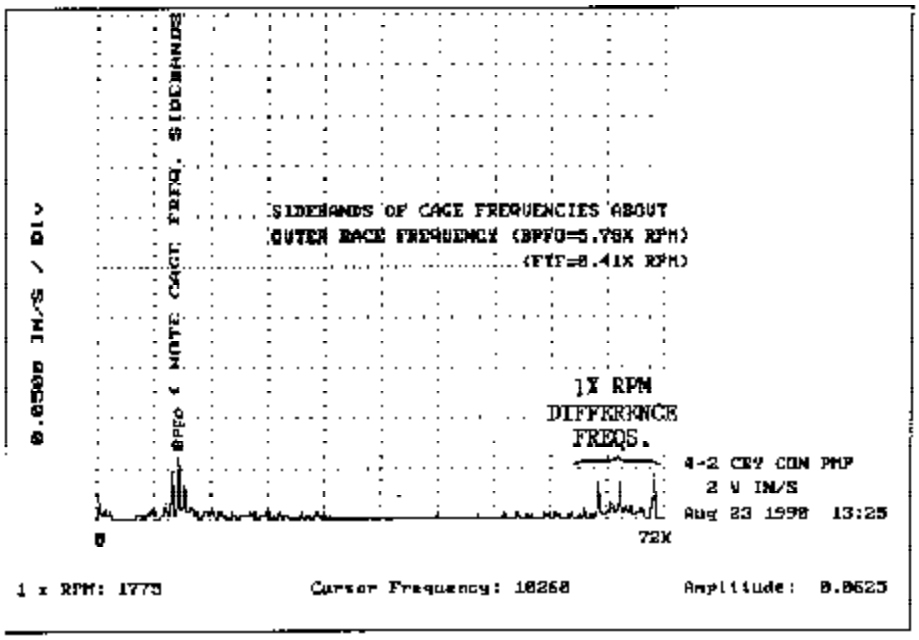
9. Ball Spin Frequency May Appear if Cage is Broken:

The appearance of a ball spin frequency does not always necessarily mean there is a defect on the rolling elements. However, it still means that there is a problem present. In this case, it can indicate that a cage is broken at a rivet and if the balls are thrusting hard against the cage (as per Reference 12).

10. Frequency Generated if More than One Rolling Element has Faults:

If more than one rolling element has defects, a frequency equal to the number of balls having defects X the ball spin frequency will be generated. In other words, if defects are present on 5 balls or rollers, a frequency at 5X Ball Spin Frequency would most often appear.

© Copyright 2001 Technical Associates Of Charlotte, P.C.



002% x 0.500 = 0.0100 1 x RPM: 1775.00

Machine	Pos	Dir	Units	Date
4-2 CRY CON PMP	2	V	IN/S	Aug 23 1990 13:25

Frequency	1 x RPM	Amplitude		Frequency	1 x RPM	Amplitude
180	0.10	0.0225		67140	37.83	0.0125
1080	0.61	0.0100		67860	38.23	0.0125
6840	3.85	0.0100		69120	38.94	0.0100
7200	4.06	0.0125		70920	39.95	0.0425
8640	4.87	0.0150				
9540	5.37	0.0475	← -FTF (.41X)			
10260	5.78	0.0625	← BPFO			
10980	6.19	0.0325	← +FTF (.41X)			
11880	6.69	0.0125				
13860	7.81	0.0100				
14580	8.21	0.0100				
21600	12.17	0.0100				
61920	34.88	0.0100				
63720	35.90	0.0350				
65160	36.71	0.0150				
65520	36.91	0.0150				
66420	37.42	0.0350				

FIGURE 6.07I
HOW THE CAGE FREQUENCY (FTF) MOST OFTEN APPEARS WITHIN A VIBRATION SPECTRUM

© Copyright 2001 Technical Associates Of Charlotte, P.C.

11. Allowable Vibration at Bearing Defect Frequencies:

It is very difficult to assign definite vibration severity levels to bearing defect frequencies similar to that which is commonly done for unbalance amplitude at 1X RPM. For one, there are a variety of rolling element bearings in a variety of different machines, each of which may provide different paths for the vibration signal to the transducer. However, one can make a statement which will apply almost uniformly for all the various types and combinations - the allowable vibration for unbalance at 1X RPM is much greater than that allowed for a defect frequency for a rolling element bearing.

Much research has proven that no absolute answer can be given to allowable vibration amplitudes at bearing defect frequencies. Not only does it depend on which particular machine a bearing is installed and what the operating speed is, it also depends greatly on which bearing failure scenario path it will travel. For example, on rare occasions, the author has personally witnessed bearings which still did not have significant damage even with an amplitude of .30 in/sec at a fundamental BPFO when this damage was concentrated at one particular location on the outer race. On the other hand, considerable damage has been discovered in other bearings when no one defect frequency had an amplitude greater than only .03 in/sec in everyday machinery like pumps and blowers running at common speeds such as 1780 RPM. In fact, extensive damage has been found in large dryer roll bearings on paper machines running less than 100 RPM with bearing frequency amplitudes that ranged from only .003 to .006 in/sec.

The key point common to each of the latter two scenarios having low vibration but significant bearing damage was that in each case, not just one bearing frequency was present in the spectra; instead, a number of bearing frequency harmonics were present (a "number of bearing frequencies present" means for example that either 3 to 5 BPFO or BPFI harmonics might simultaneously be present; or that 2 or more harmonics of BPFO might be present along with 2 or more harmonics of BPFI). In addition, further investigation has shown that when these bearing frequency harmonics were surrounded by sideband frequencies spaced at 1X RPM of the problem bearing, even more damage is indicated (particularly if these 1X RPM sidebands surround BPFO harmonics, assuming the bearing is press fit on the shaft). It should also be emphasized that these sidebands may be spaced at bearing frequencies themselves rather than at 1X RPM (i.e., 4 or 5 BPFO harmonics may all be present, each with sidebands of FTF or BSF above and below them which would likewise indicate a potentially serious problem).

Therefore, the most important thing to look for indicating significant bearing wear is the presence of a number of bearing defect frequency harmonics, particularly if they are surrounded by sidebands spaced at either 1X RPM or sidebands spaced at other defect frequencies of the bearing - independent of amplitude. If these are present in a spectrum, replace the bearing as soon as possible.

12. Evaluating Bearings on Low-Speed Machinery (less than 250 RPM):

Special precautions must be taken when making measurements on low-speed machinery. Rolling element bearings have been successfully evaluated at speeds as low as 1.5 RPM. However, one must be particularly aware of the low frequency limitations of both his analyzer and his transducer. If he uses one of the computer-loaded data collectors popular today, he should be aware that many of these instruments are outfitted with high-pass filters that begin filtering out signals below fixed frequency ranges at rates of from approximately 12 to 24 dB/ octave. This cutoff frequency may be close to DC (0 Hz) on some data collectors, but may be as high as 8 Hz (480 CPM) on other models. Thus, in the latter case, a machine may be

© Copyright 2001 Technical Associates Of Charlotte, P.C.

generating 1.00 in/sec of vibration at 240 RPM, but the instrument may display only approximately .25 in/sec at this frequency if it has such a high-pass filter. Obviously, if one attempted to make measurements at frequencies below 120 CPM with this instrument, the data collector would detect and display only a small percentage of this peak. And, if one wanted to single or double integrate the signal from acceleration to velocity or displacement, respectively, there would likely be even greater suppression of the amplitude in the data collector (also, it is important to point out that some data collectors do not integrate acceleration signals to velocity at frequencies below approximately 120 CPM; data below this cutoff frequency is left nonintegrated). In addition to filtering, the low frequency response of the analyzer is also effected by its signal conditioning and input circuitry components which also needs to be considered.

Not only does one have to be concerned about the analyzer during low frequency measurements, he also must be aware of the frequency response of the transducer whether it is an accelerometer, a velocity pickup, or a proximity probe. Typically, most general purpose accelerometers in use with data collectors today are flat within 5% between approximately 5 Hz and 10,000 Hz (300 - 600,000 CPM). However, special seismic piezoelectric accelerometers can be obtained which will extend the ±5% flat response down to as low as approximately 0.1 Hz (6 CPM). These transducers will typically have exceptionally high sensitivities ranging from 1000 to 10,000 mV/g (as compared to only 10 to 100 mV/g for the standard accelerometer) and will normally have much greater weight on the order of 400 to 1000 grams (as compared to only 10 to 50 grams for the standard unit). They also will typically have much longer discharge time constants of 20 seconds or more (compared to only 0.5 second for standard accelerometers) and will work best when connected to analyzers having input impedances of approximately 1,000,000 ohms. Finally, the low frequency performance of these transducers can be further enhanced by connecting them to special power supplies and signal conditioning equipment.

In addition to the instrument limitations discussed, it should be pointed out that both 1X RPM and bearing frequency amplitudes themselves will inherently be much lower on large, low-speed machinery. In the case of 1X RPM, there will be minimal unbalance forces since these vary with the square of speed. Thus, if one attempts to specify spectral alarm bands for this machinery, he will have to spec them at much lower alarm amplitudes than for machinery rotating above 1200 RPM. The best way to specify them will be to capture actual data and perform statistical analysis on the overall levels and those of individual frequency bands.

Fortunately, even though many of today's standard data collectors and transducers may not be capable of evaluating 1X RPM and 2X RPM vibration on much low-speed machinery, these same instruments might still be able to satisfactorily evaluate the health of their bearings. Please refer to Figure 6.07G which shows some typical rolling element bearing frequencies. Even though the fundamental cage (FTF) and ball spin frequencies (BSF) may still be below the "reach" of the system, the more common outer race (BPFO) and inner race frequencies (BPFI) will most often range from 4 to 12X RPM. Therefore, a measurement system which could not "see" vibration at, for example, 100 RPM speed very likely could detect fundamental outer and inner race frequencies which would probably range from about 500 to 1000 CPM on this machine with little or no loss of signals. And, of course, harmonics of these bearing frequencies could easily be evaluated.

Each of these items will be further discussed in more detail in the seminar entitled "Analysis II". For now, the most important point is that one can successfully evaluate bearing health on low-speed machinery if he takes into account the frequency response performance characteristics of his instrumentation system and takes the necessary provisions outlined in this section.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

13. Where Transducer Should be Placed to Properly Detect Bearing Frequencies:

It is most important to place the vibration transducer as close as possible to the load zone of the bearing, particularly if the bearing only supports radial load. Reference 13 points out that this is of critical importance for a spherical roller bearing in which he claims the vibration signal strength can be affected by as much as 100% if the transducer is not placed in the load zone, giving the analyst a false sense of security when he actually does have a problem.

14. Approximate Bearing Defect Frequencies if only the Number of Rolling Elements is Known:

Figure 6.07F provides 4 formulas which approximate the bearing defect frequencies if only the number of rolling elements (N_b) is known (as per Reference 18). These equations have been proven to be considerably more accurate than older approximations which assumed BPFO to equal approximately .4 x N_b x RPM and BPFI to equal approximately .6 x N_b x RPM. Still, optimum precision can only be realized if one knows each of the other bearing design parameters (B_d, P_d and Ø), particularly in the case of the ball spin frequency (BSF).

15. Sensing of Improper Bearing Load or Installation:

Not only can bearing frequencies be used to detect defects within the bearings, but also they can be used to detect when bearings are improperly loaded or installed **(that is, they do not always indicate that a defect is present within the bearing)**. For example, even when a new bearing is installed, if there is excessive interference of the bearing seat on the bearing housing causing it to be "jammed into" the seat, it can result in takeup of all the internal clearances forcing the rolling elements against the races. If this occurs, the bearing will immediately generate ball pass frequencies of the outer and/or inner race upon startup. In addition, the author has taken data several times when thrust bearings were installed backwards. When this has occurred, the improperly installed thrust bearings have generated excessively high amplitudes at race defect frequencies, sometimes on the order of 1.0 in/sec, or greater. Excessive press fit of a bearing onto a shaft can also immediately generate a pronounced outer race or inner race defect frequency (BPFO or BPFI), letting the analyst know excessive and/or improper load has been placed on the bearing. And, even though no real wear might yet have occurred, if the problem is not detected and corrected, the bearing will likely fail long before its predicted life. In all improperly loaded situations, if the same bearings were re-installed properly or loaded properly, the defect frequencies were significantly reduced or disappeared altogether. This has happened on a number of occasions. In fact, one client manufacturing textile machinery uses this technique to detect assembly problems in its quality assurance program.

16. Frequencies Generated by Inadequate Bearing Lubrication:

Reference 12 states that "unique signatures generated by inadequately lubricated bearings are characterized by 3 or 4 peaks in the frequency range of 900 to 1600 Hz (54,000 to 96,000 CPM). The difference frequency between the peaks ranges from 80 to 130 Hz (4800 to 7800 CPM). Some signatures of properly lubricated bearings contain these frequency components; however, the amplitude is very low - about 0.05 in/sec or less. The amplitude increases to as much as 0.10 or 0.20 in/sec when the lubrication is inadequate". He adds that "empirical evidence indicates that frequencies from 900 to 1600 Hz are natural frequencies of the installed bearing." With this in mind, if frequencies within this range (approximately 50,000 to 100,000 CPM) do occur with difference frequencies on the order of 5000 to 8000 CPM, it may be a good idea to check lubrication, particularly if high spike energy (or equivalent) levels are also measured on this same bearing housing. Also, even

© Copyright 2001 Technical Associates Of Charlotte, P.C.

greater evidence of lubrication problems would occur if neither the high spike energy nor a spectrum having these components was present on the other bearing that is supporting this same shaft.

In addition, some research has shown that inadequately lubricated bearings which allow metal-to-metal contact can also generate bearing defect frequencies. In these cases, one might go ahead and add lubricant while monitoring both overall spike energy (or equivalent) and vibration FFT spectra to determine if the ultrasonic levels drop considerably, and to find if the bearing defect frequencies disappear from the spectrum. If they do, he should return to the machine 12 to 24 hours later and see if either have reoccurred. If not, lubrication was likely the problem. If they do return, the bearing is likely suffering a wear problem as described in Section 6.073.

(d) Sum and Difference Frequencies:

When a single defect has developed on a bearing, it will generate a defect frequency and will begin deteriorating. When the defect grows, it can contribute to the development of other defects in the bearing. In so doing, other frequencies will be generated and a number of patterns might develop. Some frequencies will add to and subtract from others. In fact, the fundamental frequency for a particular defect might never occur. When it appears, this defect frequency might act as a sideband to other frequencies which are already present. For example, a cage frequency itself will not normally occur at its fundamental frequency of approximately .35X to .45X RPM. Instead, the cage frequency itself normally will sideband the race frequencies (BPFO or BPFI) or the ball spin frequency of the bearing (see Figure 6.07I). In some cases, bearing defect frequencies can even modulate frequencies generated by sources other than bearings. Following below are some of the more important facts concerning sum and difference frequencies and their behavior:

1. It is not uncommon for the fundamental ball spin frequency or harmonics never to appear even if significant faults are present on the rolling elements. In this case, BSF will appear, but as a sideband of other frequencies.

2. If only a single fault is present on either the outer or inner race, only a single race frequency will appear. However, when faults begin to appear around the periphery, a number of harmonics of these race frequencies will appear.

3. When defects grow on raceways, the amplitude of the race frequencies themselves will often increase somewhat. However, even greater indicators of deterioration are the number of bearing frequency harmonics, as well as the appearance of 1X RPM sideband frequencies above and below the race frequencies (particularly BPFO). In effect, the unbalance forces at shaft speed will tend to modulate the frequency components of the races, generating sidebands. In the case of BPFI, 1X RPM sidebands are often created around it since its amplitude is modulated with much greater response when the inner race defect impacts rolling elements within the load zone than those generated outside the load zone. The rate of this modulation in amplitude is at 1X RPM, therefore generating the 1X RPM sidebands (see Figure 6.07H).

4. In rolling element bearing frequency analysis, the emphasis is <u>not on amplitude</u>, but on <u>content of the spectra</u>. In fact, <u>amplitudes of bearing frequencies themselves often begin to drop as condition worsens</u>, particularly in the case where faults begin to propagate around the periphery of the outer or inner race. In this case, serious weight should be placed on the fact that a multiple number of fault frequencies are appearing and that many of these fault frequencies are sidebanded by vibration at bearing RPM.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

5. As deterioration continues, modulation effects can continue to have more and more influence until eventually, the spectrum can become a series of nothing but 1X RPM harmonics.

6. Reference 14 shows that double row bearings having staggered rolling elements can generate 2 sets of bearing defect frequencies - (a) If a defect is confined to only one side of the raceway, the bearing defect frequencies generated should be calculated using the number of rolling elements only in that single row; however, (b) if a defect occurs on both sides of the raceway, the frequencies should be calculated using the total number of rolling elements. Here again, either of these frequencies can act as a sideband around other frequencies with the sideband difference depending on whether the fault has occurred on one or both raceways. Importantly, this number of balls (either those for one row or the total number) must be entered into the outer race, inner race and ball spin frequencies.

7. A single defect on the inner race of a rolling element can be difficult to diagnose from frequency spectra alone since amplitudes will most often be low, and discrete spectral lines might not be seen at calculated defect frequencies. This particularly applies to bearings with inner race rotation mounted with an interference fit between the inner race and the shaft. In these cases, the load zone might be noticeably less than 180° which explains why discrete peaks at defect frequencies may not appear. Reference 15 points out the real problem occurs if a crack appears on the inner race. In these cases, the shaft fit can be relieved and internal clearances of the bearing lost. He points out that such a situation is extremely dangerous which can bring about rapid seizure of the bearing causing the inner race to spin on the shaft or the outer race to spin in the housing resulting in catastrophic failure. In these cases, since the inner race frequency itself or harmonics may not show, but act as a sideband on other frequencies, he recommends referring to the time domain (or time waveform) in addition to the vibration spectra themselves.

8. In rolling element bearings, when multiple running speed harmonics are present, they can signal either looseness of the bearing on the shaft or within its seat or, much more importantly, they can likewise signal a bearing turning on the shaft or in the housing. The author has been involved in several situations where a number of very low amplitude running speed harmonics (less than .04 in/sec) were present which resulted in catastrophic failures, even with only the low amplitudes. In many of these cases, there were in fact no real defects within the bearings themselves, but the bearing turning on the shaft resulted in severe shaft damage (in one case, over .25 inch diameter was lost before eventual failure).

Thus, there is great information present within the signatures of rolling element bearings that will help identify their current condition and assist the analyst in recommending possible corrective actions. Following in Section 6.073 will be a presentation of a series of typical spectra which can be used by the analyst to track the condition of rolling element bearings under some of the most common failure scenarios which have occurred in the experience of the author.

6.073 Typical Spectra For Tracking Failure Stages Through Which Rolling Element Bearings Pass:

During the last decade, concentrated research and experimental investigations have been conducted on how to best evaluate rolling element bearing condition using vibration analysis and high frequency enveloping techniques. Some of the better papers which have been written include References 11, 12 and 16. In addition, Technical Associates has been deeply involved in further developing rolling element bearing diagnostic capabilities. We have consulted with a variety of clientele on a wide array of machinery ranging from massive, low-speed machines outfitted with large, expensive rolling element bearings to very small, high speed rolling element bearing machinery.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

Experiments have been conducted purposely running rolling element bearings to failure while carefully tracking them with vibration signatures (Reference 11). At various intervals, the bearings were broken open and closely inspected for the extent of damage, the type of deterioration and the probable cause for the onset of faults within the bearings. These results would then be closely correlated with the vibration spectra, along with various ultrasonic frequency measurements (i.e., Spike Energy, Shock Pulse and HFD). The bearings were purposely opened up and examined when such data indicated them to be in a wide range of condition (from those with supposedly no faults to those indicated to have extensive damage). Some only had increases in ultrasonic measure-ments (not in vibration); others showed increases only in high frequency regions; while clearly defined bearing defect frequencies were within vibration spectra of others (some were allowed to catastrophically fail while vibration and ultrasonic responses were closely tracked). Such experiments were performed on a variety of rolling element bearings at different speeds and conditions. This work has enabled the development of fairly well proven diagnostic methods for the tracking of rolling element bearing condition using vibration signature analysis, along with time waveform and ultrasonic analysis.

It was found that the majority of rolling element bearings followed a fairly predictable failure path from the very onset of deterioration through eventual catastrophic failure. This failure path is graphically portrayed in Figure 6.07J which plots bearing damage versus time. Note importantly that bearing damage typically will increase exponentially during the final 10% to 20% of its life. It is here where intensive research and field investigations have found how to use vibration analysis and high frequency enveloping tools to identify "failure stages". From this, a classic 4 stage failure scenario has been developed which will apply to approximately 80% of rolling element bearing failures. This will be presented in Figure 6.07K as Scenario A entitled "4 Primary Failure Stages Through Which Most Rolling Element Bearings Pass". Text describing each of the 4 failure stages will accompany Figure 6.07K.

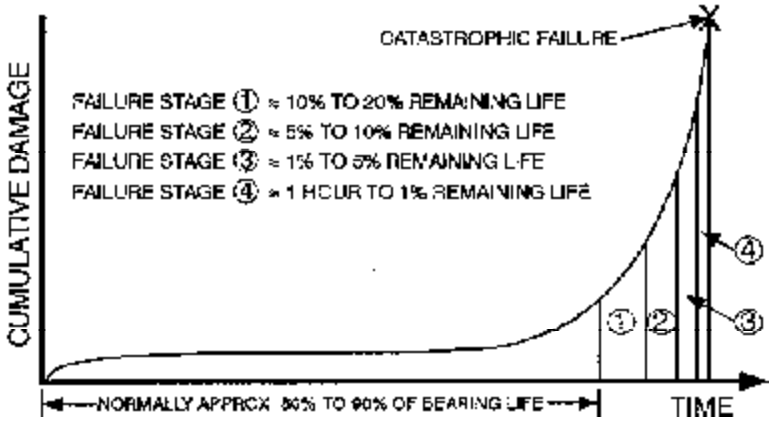


FIGURE 6.07J
TYPICAL FAILURE PATH TAKEN BY ROLLING ELEMENT BEARINGS

As was expected, not all rolling element bearings follow a failure path identical to that which will be outlined for Scenario A. At least five (5) other failure scenarios have been identified. However, due to considerations limiting article length, only this first scenario encompassing the normal failure path taken by approximately 80% of rolling element bearings will be covered (Scenario A). The complete paper referenced in notes accompanying the article covers the remaining five failure scenarios and gives real-world examples of each (Reference 26).

© Copyright 2001 Technical Associates Of Charlotte, P.C.

SCENARIO A. 4 PRIMARY FAILURE STAGES THROUGH WHICH MOST ROLLING ELEMENT BEARINGS PASS (FIGURE 6.07P):

Figure 6.07K presents typical velocity spectra for each of the 4 failure stages for most rolling element bearings. These spectra follow the bearing from the very onset of bearing problems in Stage 1 through imminent failure of the bearing in Stage 4 (see Figure 6.07J). Note that the overall spike energy (or HFD) amplitudes given to the right of each spectrum are meant to be rough approximations only. Also, note that documentation below the title of each one of the stages refers to an approximate "L_{10} Life Remaining" which corresponds to the approximate remaining anticipated life of the bearing based on a 90% confidence level (L_{10} Life is documented and its formula is given in the introductory section of this Section 6.07). Today, in general, the machine designer normally will attempt to provide a design life of approximately 5 to 10 years for most common machinery. Therefore, when a percentage on the order of 10% of L_{10} Life is quoted below, this will generally mean from 6 months to 1 year remaining life anticipated (depending on the type of machine and its intended application). Of course, this can vary widely with the machine type, with particular design parameters and whether or not the bearings are provided with the proper lubrication, operating temperature and subjected to acceptable vibration levels.

It is important to point out that bearings do not follow a linear deterioration path, but instead tend to fail exponentially; that is, when the bearing for example enters Stage 3 outlined below, it may still have a fair remaining life, or it may fail rapidly. Once the bearing enters Stage 3, the failure can progress rapidly, particularly towards the end of this stage. Generally, low speed bearings may still have fairly predictable remaining life (unless exposed to high dynamic loads - see L_{10} Life Formula discussion in the introduction to this Section 6.07 which shows that bearing life is inversely proportional to the 3rd power of imposed loading).

Table 6.07A compiled by Charles Berggren in Reference 16 (reprinted here with his permission) roughly adheres to the 4 stages illustrated in Figure 6.07K and documented below:

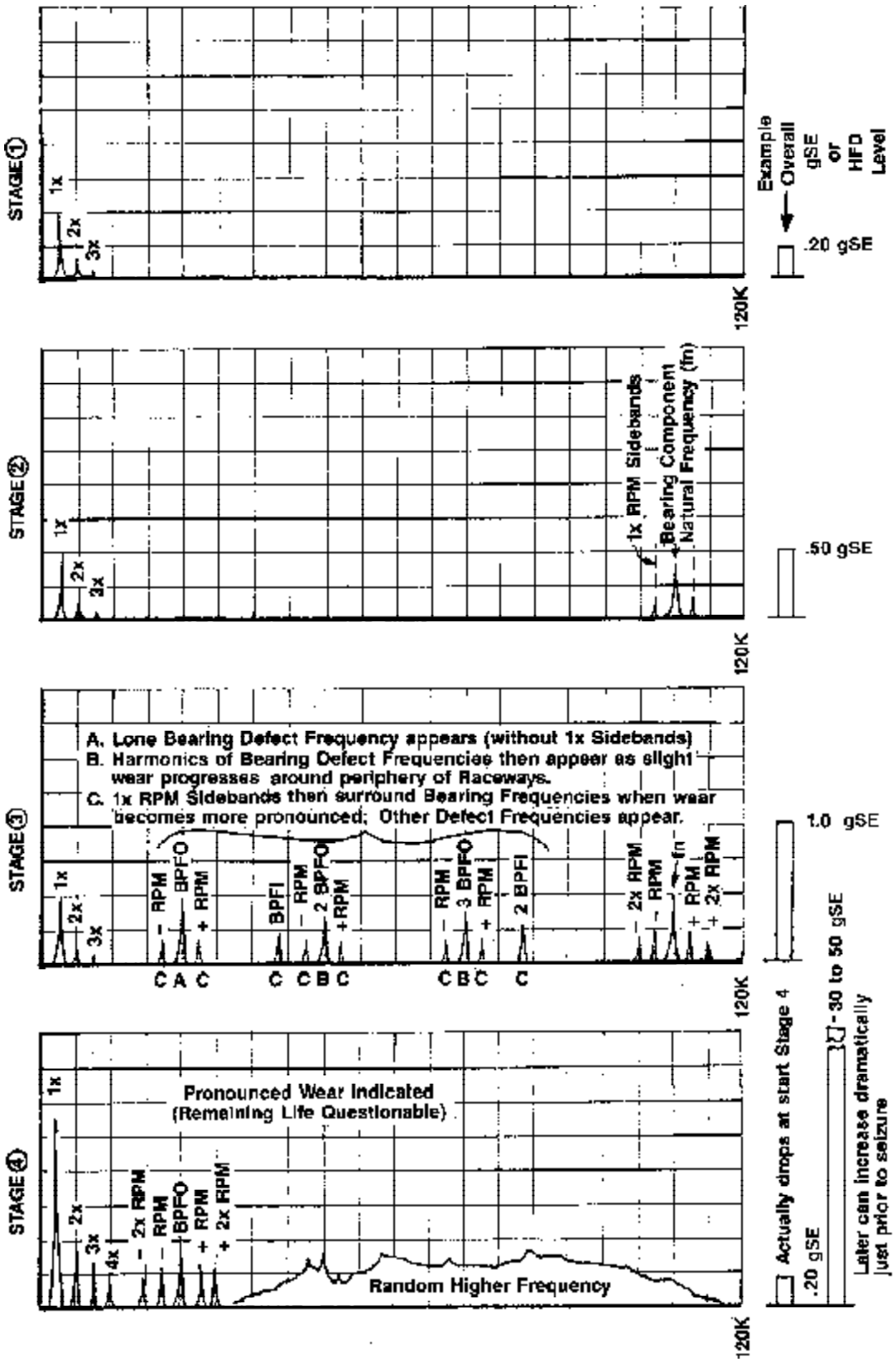
Stage 1 (Approximately 10% to 20% L_{10} Life Remaining):

The appearance of Spike Energy (or HFD, Shock Pulse, etc.) from very low levels (below .02 to .04 gSE) up to .15 to .25 gSE is the event which defines the onset of Failure Stage 1. For example, Stage 1 shows a normal spectrum indicating a healthy bearing and has only the normal first 3 running speed harmonics in the velocity spectrum. The only evidence of possible bearing problems is that Spike Energy has grown from near 0 to approximately .20 gSE (this is an example amplitude only; actual levels depend on the particular bearing and how close the measurement is to the bearing housing). Note that the Spike Energy itself is not meant to be part of the spectrum in Figure 6.07K but only an overall level (Spike Energy spectra are also available and can be used to confirm that a bearing problem is the cause for increase of this Spike Energy overall level. The Spike Energy response itself responds in the ultrasonic frequency ranges in the region of 2,000,000 CPM (approximately 35,000 Hz). HFD is also a high frequency overall level extending as high as 1,800,000 CPM (30,000 Hz).

During Stage 1, no sound will be detectable by the human ear indicating bearing damage and no change in bearing temperature would be anticipated at this point. Table 6.07A shows that no defects detectable by the human eye would be expected in this stage since such defects would be near microscopic at this stage. Even though physical examination by the naked eye and hand would not normally indicate wear at this stage, a metallurgical analysis would likely reveal damage within the outer .004 to .006 inch (.10 to .15 mm) surface layer of the races and/or rolling elements.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

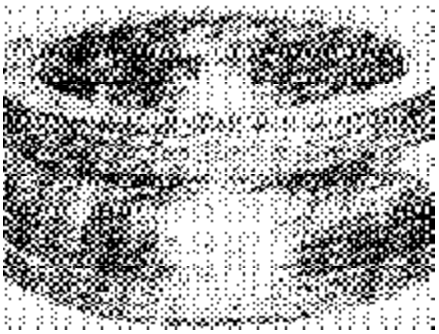
FIGURE 6.07K
4 PRIMARY FAILURE STAGES THRU WHICH MOST ROLLING ELEMENT BEARINGS PASS (VELOCITY SPECTRA)



© Copyright 2001 Technical Associates Of Charlotte, P.C.

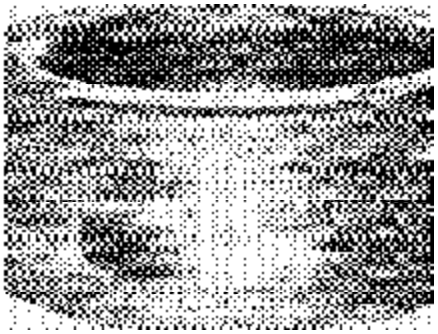
TABLE 6.07A
4 ROLLING ELEMENT FAILURE STAGES (Ref. 16)

a. Initial Phase.



- noise level normal
- temperature normal
- measurable increase in ultrasonic sound, acoustic emission, spike energy, and outer race deflection
- overall vibration low; no discrete spikes at bearing
- frequencies remaining life more than 10% of B-lO rating*

b. Second Phase.

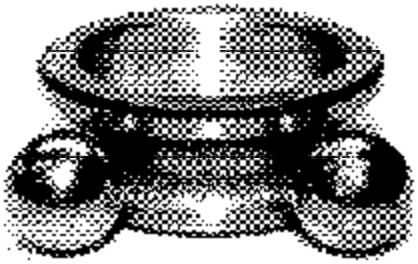


- slight increase in noise level
- temperature normal
- large increase in ultrasonic sound, ucoustic emission, spike energy, and outer race deflection
- slight increase in overall vibration acceleration and velocity
- bearing frequencies clearly visible on log scale, barely visible on linear scale of vibration spectrum; noticeable rise in noise floor
- remaining life less than 5% B-10 rating

c. Third Phase.

- noise level quite audible
- slight increase in temperature
- very high ultrasonic sound, acoustic emission, spike energy,and outer race defiection
- large increase in overall vibration acceleration and velocity
- bearing frequencies with harmonics and sidebands clearly visible on linear scale
- of vibration spectrum; noticeable rise in noise floor
- remaining life less than 1% B-10 rating*

d. Final Phase.



- change in pitch of noise level
- significant temperature increase
- gradual decline followed by rapid increase in ultrasonic sound, acoustic emission, spike energy, and outer race defiection immediately prior to failure
- significant increase in overall vibration displacement and velocity; decrease in acceleration

© Copyright 2001 Technical Associates Of Charlotte, P.C.

Stage 2 (Approximately 5% to 10% L_{10} Life Remaining):

Slight bearing defects begin to "ring" natural frequencies of the installed bearing components (see Figure 6.07K). These natural frequencies are concentrated in the 30,000 to 120,000 CPM range (500 to 2000 Hz). Natural frequencies of most bearings lie between 30,000 and 90,000 CPM (500 - 1500 Hz). These are natural frequencies of the assembled rolling element bearings themselves which do not change in frequency with a change in operating speed (however, they normally will show higher amplitudes with increasing speed due to greater impact velocity). These natural frequencies are excited by the momentary impact between the defective rolling elements and bearing races which not only excite the bearing natural frequencies, but also increase Spike Energy or HFD response (for example, roughly doubling in many cases). It has been the experience of the author that during initial Stage 2, only one or more frequencies appear in these higher frequency regions. Later, towards the end of Stage 2, these frequencies will not only grow in amplitude, but also become modulated with the running speed when wear progresses (that is, 1X RPM sidebands will later appear above and below these natural frequencies). Although modulation of these bearing component natural frequencies most often occurs at 1X RPM, it should be pointed out that such sidebands can also be spaced at bearing defect frequencies (BPFO or BPFI) about the bearing natural frequencies.

Note that the defects themselves may not yet be readily visible to the naked eye in Stage 2. There should be only a slight increase in bearing noise and its temperature should still be roughly normal. Notice that bearing defect frequencies will not likely yet be visible in the velocity spectrum. However, acceleration spectra may now begin to pick up harmonics of such bearing defect frequencies, particularly if a log amplitude scale is employed. Still, at this stage, bearing defect frequency response will normally be erratic.

Stage 3 (Approximately 1% to 5% L_{10} Life Remaining):

Note that each of 3 progressive events are documented for Stage 3 in Figure 6.07K (shown as A, B and C). The letters shown below the horizontal axis of the Stage 3 spectrum correspond to peaks which appear during these 3 progressive events. For the first time, bearing defect frequencies associated with faults in the inner race (BPFI), outer race (BPFO), rolling elements (BSF) and cage (FTF) appear in the velocity spectrum (Event A). Later on in Stage 3, harmonics of these bearing defect frequencies then will appear as slight wear progresses around the periphery of the raceways and/or faults appear on more than one rolling element (Event B). Normally, outer race (BFPO) frequencies themselves will not be modulated by running speed when they first appear. However, when deterioration progresses, 1X RPM sidebands will surround the outer race frequencies. In the case of inner race frequencies (BFPI), 1X RPM sidebands will often appear as soon as do the inner race frequencies (or their harmonics) since inner race defects will have much more response when they are impacted by rolling elements and are within the load zone (they will see very little response when such impacts occur with the inner race defect outside the load zone), Therefore the 1X RPM sidebands often appear with BPFI (and harmonics) due to what is known as "amplitude modulation" (which is covered in depth in the "Analysis II" seminar).

Spike Energy and HFD will continue to grow, doubling or tripling in amplitude. At the end of Stage 3, not only will more and more 1X RPM sidebands appear around bearing defect frequencies, but more sideband families will also usually appear around the bearing component natural frequencies (Event C).

© Copyright 2001 Technical Associates Of Charlotte, P.C.

A word of caution should be taken at this point. When the bearings approach the conclusion of Stage 3, the rate of wear becomes highly unpredictable. In any case, it will be well into the exponential part of the failure curve. How much longer the bearing lasts will largely depend on its lubrication, temperature, cleanliness and dynamic loads being imposed upon it by vibration forces from unbalance, misalignment and so forth. At this point, there will be a noticeable change in sound level and frequency and a slight increase in bearing housing temperature. Addition of lubricant at this point may temporarily lower Spike Energy and possibly have some effect in reducing vibration. However, since wear is the problem at this point, both vibration and Spike Energy will return, normally within 12 to 24 hours (the lubricant addition at this point could actually hasten failure if the bearings already were over-lubricated).

Figures 6.07L and 6.07M show a spectrum and trend plot taken from a pump bearing with a classic Stage 3 bearing failure underway. Notice that the Spike Energy increased from .271 to .944 gSE between May 24 and August 23 measurements exceeding the alarm of .750 gSE (Figure 6.07M). Then, looking at the velocity spectrum of Figure 6.07L, each of 4 harmonics of inner race frequency are present (BFPI = 7.44 X RPM = 13,200 CPM). Note the harmonics at 26400, 39600 and 52800 CPM. Also note the multiple 1X RPM sidebands which have appeared around 2X BPFI and 3X BPFI. Looking at both the velocity spectral data and Spike Energy overall level, the plant would be well advised to replace this bearing right away even though amplitudes of all bearing frequencies are below .10 in/sec.

Stage 4 - Approaching Catastrophic Failure
(Approximately 1 Hour to 1% L_{10} Life Remains):

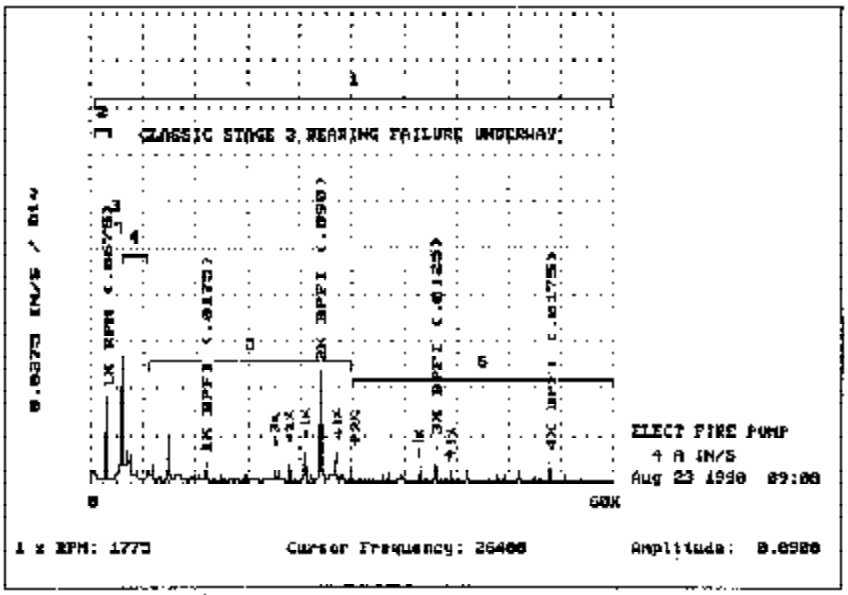
The level of 1X RPM normally begins to grow for the first time throughout the bearing failure process in Stage 4, along with harmonics at 2X and 3X RPM. Spike Energy and HFD levels actually begin to drop and amplitudes of the higher bearing frequency harmonics and natural frequencies also will normally drop. Many 1X RPM sidebands appear around bearing defect frequencies (indicating pronounced wear throughout the periphery of the bearing and/or load zone). There will now be a noticeable change in pitch of the bearing noise and likely a significant increase in bearing housing temperature.

Later on in Stage 4, discernible bearing defect component natural frequencies actually begin to disappear and are replaced by a random high frequency "noise floor" which can extend far down into the spectrum obliterating discrete frequency peaks.

There may now be one week of remaining bearing life, on the other hand, the bearing may fail within one hour! The point is that no one knows! Finally, at the end of Stage 4, Spike Energy and HFD normally will decline again (as the bearing itself "disappears"); but, just prior to failure, Spike Energy levels can grow dramatically up to 50 to 100 gSE just before final seizure. Thus, a bearing should never be allowed to continue to operate in Stage 4, for no one knows when it will catastrophically fail.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

FIGURE 6.07L



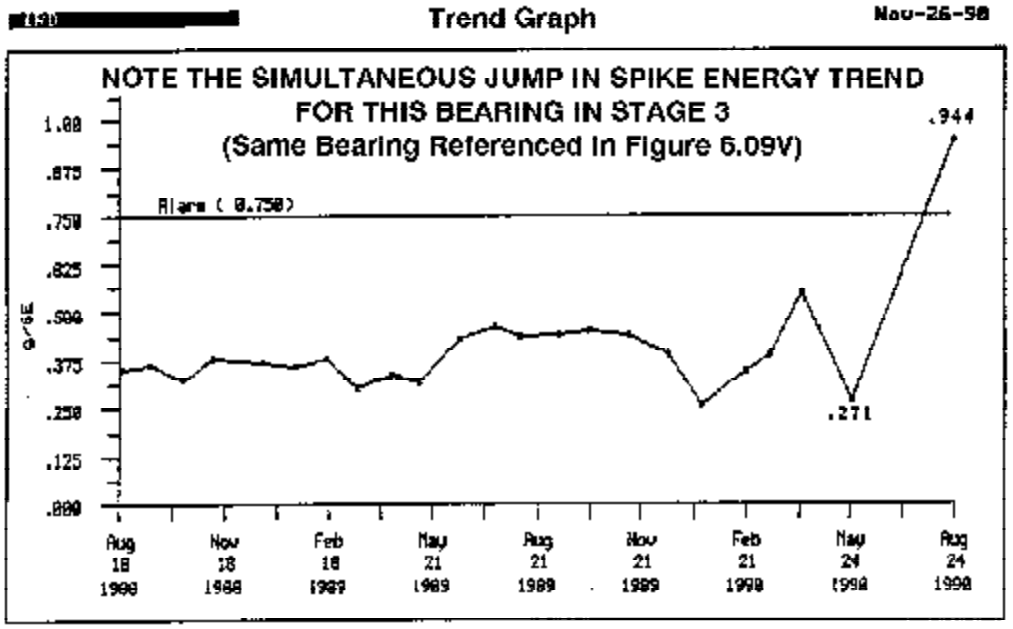
2% x 0.500 - 0.0100 1 x RPM: 1775.00

Machine	Pos	Dir	Units	Date
ELECT FIRE PUMP	4	A	IN/S	Aug 23 1990 09:08

Frequency	1 x RPM	Amplitude	Frequency	1 x RPM	Amplitude
150	0.08	0.0100	39600	22.31	0.0275←3xBPFI
600	0.34	0.0100	41400	23.32	0.0100
1800	1.01	0.0675	43200	24.34	0.0100
3600	2.03	0.1000	52800	29.75	0.0175←4xBPFI
4050	2.28	0.0250			
4500	2.54	0.0200			
6600	3.72	0.0100			
7200	4.06	0.0150			
8850	4.99	0.0350			
13200	7.44	0.0175←BPFI			
21000	11.83	0.0100			
22800	12.85	0.0150			
24600	13.86	0.0225			
26400	14.87	0.0900←2xBPFI			
28200	15.89	0.0225			
30000	16.90	0.0125			
37800	21.30	0.0125			

© Copyright 2001 Technical Associates Of Charlotte, P.C.

FIGURE 6.07M



ELECT FIRE PUMP 4 H .750 g/SE

IRD Mechanalysis Data Analysis System

Machine ELECT FIRE PUMP Pos 4 Dir H Units g/SE Alarm .750

Ampl	Insp Code	Date mm/dd/yy	Time	Ampl	Insp Code	Date mm/dd/yy	Time
.353	0	08-16-88	9:25	.442	0	09-07-89	8:51
.361	0	09-13-88	0:07	.449	0	10-05-89	8:50
.320	0	10-11-88	9:18	.442	0	11-09-89	10:00
.378	0	11-07-88	9:19	.390	0	12-14-89	15:38
.368	0	12-20-88	10:34	.261	0	01-11-90	8:52
.355	0	01-17-89	9:10	.347	0	02-19-90	8:40
.380	0	02-14-89	8:51	.188	0	03-12-90	8:58
.302	0	03-14-89	8:45	.549	0	04-09-90	7:54
.338	0	04-13-89	8:03	.271	0	05-24-90	7:51
.316	0	05-09-89	9:18	(.944)	0	08-23-90	9:08
.429	0	06-13-89	9:17				
.460	0	07-13-89	9:20				
.434	0	08-03-89	9:25				

Page 1

Machine Report (Latest Data) for: ELECT FIRE PUMP

Date	Time	Pos	Dir	Ampl	Units	Change	Alarm	Inspection
Aug-23-90	09:06	1	A	.150	IN/S	-6%	.300	
Aug-23-90	09:06	1	H	.129	IN/S	48%	.300	
Aug-23-90	09:06	1	H	.329	g/SE	186%	.600	
Aug-23-90	09:06	1	O	.117	IN/S	-----%	.300	
Aug-23-90	09:06	1	V	.077	IN/S	8%	.300	
Aug-23-90	09:07	2	A	.167	IN/S	-16%	.300	
Aug-23-90	09:07	2	H	.127	IN/S	14%	.300	
Aug-23-90	09:07	2	H	.332	g/SE	-1%	.600	
Aug-23-90	09:07	2	O	.129	IN/S	-----%	.300	
Aug-23-90	09:07	2	V	.061	IN/S	-65%	.300	
Aug-23-90	09:07	3	A	.211	IN/S	41%	.300	
Aug-23-90	09:07	3	H	.180	IN/S	10%	.300	
Aug-23-90	09:07	3	H	.337	g/SE	42%	.600	
Aug-23-90	09:07	3	V	.189	IN/S	26%	.300	
Aug-23-90	09:08	4	A	.204	IN/S	-19%	.300	
Aug-23-90	09:08	4	H	.204	IN/S	2%	.100	
Aug-23-90	09:08	4	H	*.944	g/SE	(248%)	.750 ←	
Aug-23-90	09:09	4	V	.121	IN/S	-15%	.100	

18 lines this report

© Copyright 2001 Technical Associates Of Charlotte, P.C.

Other Rolling Element Bearing Failure Scenarios Which Have Been Identified

While Scenario A outlined in Figure 6.07K probably encompasses the failure paths followed by approximately 80% of rolling element bearings, other scenarios have been identified. To date, five (5) other failure scenarios have been documented. Due to limitations on the length of this article, it is not feasible to provide all the text, spectrum diagrams and example illustrations as was done with Scenario A for these other scenarios (complete documentation on each of the remaining failure scenarios is included within the "Analysis II" seminar text, Reference 26). However, following below in Table 6.07B is a tabulation of each of these remaining failure scenarios:

TABLE 6.07B
DESCRIPTION OF OTHER IDENTIFIED ROLLING ELEMENT BEARING FAILURE SCENARIOS

FAILURE SCENARIO	SCENARIO DESCRIPTION
B.	"One Pronounced Localized Fault on One Raceway Exciting One Dominant Bearing Defect Frequency"
C.	"Continual Wear Throughout the Periphery of a Raceway"
D.	"Development of a Serious Fault Frequency Acting as a Sideband Rather than a Fundamental Frequency"
E.	"Condition Deterioration Ending Either With Severe Mechanical Looseness or the Bearing Turning on the Shaft"
F.	"Development of Excessive 1X RPM Modulation About Race Frequencies Ending up with Excessive Fatigue Spalling on Raceways"

© Copyright 2001 Technical Associates Of Charlotte, P.C.

6.08 INTRODUCTION TO GEAR PROBLEM DETECTION

Following in this paper will be an introduction on how problems are detected with gears using vibration signature analysis (occasionally with the assistance of ultrasonic spectral analysis such as spike energy spectra). This paper will cover the fundamentals of gear analysis, particularly concentrating on the wear of gear teeth. The focus of this paper will be on how wear of gear teeth is detected while the emphasis of papers in higher level seminar texts will be on how to detect cracked or broken gear teeth, gear eccentricity and/or excessive backlash, gear assembly phase problems, hunting tooth problems, etc. However, it is of great importance that the analyst who is beginning to evaluate gears understand the significance of gear mesh frequencies (#teeth X RPM) and of sideband frequencies surrounding the gear mesh frequencies (spaced at 1X RPM on either side of gear mesh frequencies).

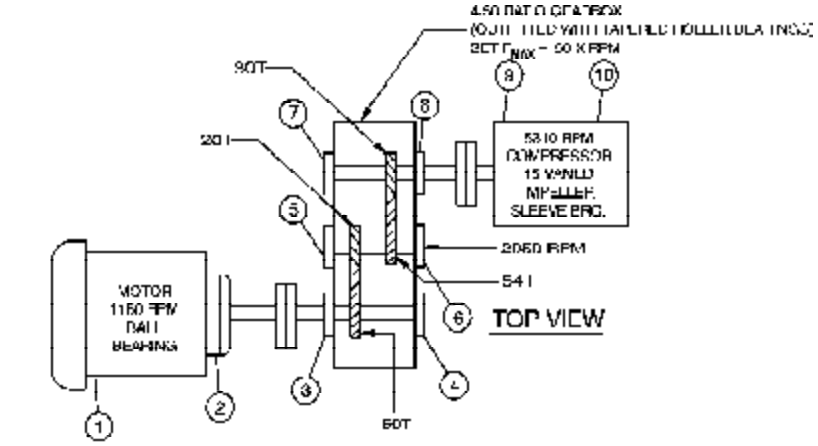
Figure 6.08A will help illustrate how this frequency important to gear analysis known as the gear mesh frequency (GMF) is calculated. As the figure shows, GMF = #Gear Teeth X Gear RPM. In this case, notice there are two gear mesh frequencies in this 2-stage gearbox (GMF_1 and GMF_2). The first mesh involves a 50 tooth gear operating at 1180 RPM meshing with a 20 tooth gear at 2950 RPM. The next mesh (GMF_2) involves a 54 tooth gear operating at the same 2950 RPM meshing with a 30 tooth gear running at 5310 RPM. Note just below the drawing the calculations of GMF_1 and GMF_2. Notice that the number of teeth and RPM on either gear can be used to calculate the gear mesh frequency. For example, in the case of GMF_1, it can be calculated by multiplying 50T X 1180 RPM, which is the same as 20T X 2950 RPM, or 59,000 CPM.

6.081 Specification of Spectral Setup for Detecting Gear Wear:

Later, it will be pointed out that one should always use a frequency span up to a minimum of 3.25 X the gear mesh frequency when evaluating gears (if GMF is unknown, it is probably a good idea to specify the maximum frequency (F_{MAX}) at about 200X RPM and to employ 1600 FFT Lines to allow detection of low-speed bullgear sideband frequencies as well as higher speed pinion sidebands). This is due to the fact that many gear wear problems do not necessarily occur at the fundamental gear mesh frequency (1X GMF), but even at 2X GMF or 3X GMF. In fact, one of the most common frequencies at which gear wear is first detected is at 3X GMF. This is likely due to the fact that each mesh event includes three separate events - a sliding action as one tooth enters mesh with another; a rolling action as they approach the root of the mating gear; and another sliding action as the teeth disengage. Any interruption to this meshing action can therefore generate frequencies at three pulses per mesh (or 3X GMF). Since taking data up to 3.25X GMF will often require high frequency measurements, it will often mandate acceleration spectra if F_{MAX} exceeds approximately 240,000 CPM (or 4000 Hz). In addition, use of 1600 line FFT spectra is necessary in order to even detect lower speed bullgear RPM sidebands at these high frequencies. Using 1600 lines of FFT resolution will separate them from the GMF fundamental or harmonics.

Because gear analysis requires measurements up to 3.25X GMF as a minimum, it may sometimes require the use of two separate transducers on each gearbox position. The first transducer might be used to measure a span extending up to only about 40X or 50X RPM which would detect such problems as unbalance, misalignment, eccentricity, bearing wear, etc. The second measurement at the same gear locations (and likely with a much higher frequency span) would be used to evaluate potential gear wear. Therefore, each individual location may require two separate measurements having two different frequency spans for complete evaluation of gear health during each survey.

© Copyright 2001 Technical Associates Of Charlotte, P.C.



GMF_1 = (50T)(1180 RPM) = (20T)(2950 RPM)

GMF_1 = 59,000 CPM

POS. 3HI F_{MAX} = GMF X 3.25

191,750 CPM

USE POS 3HI F_{MAX} = 200,00 CPM

GMF_2 = 54T X 2950 RPM = (30T)(5310 RPM)

GMF_2 = 159,300 CPM

POS. 6HI F_{MAX} = 3.25 X 159,300

517,725 CPM

USE POS 6HI F_{MAX} = 540,00 CPM

(Must Measure Acceleration Here)

NOTE:
Spectral Band Alarm Setup for the 2-Stage speed increaser shown in Figure 6.08A is shown for all motor, gearbox and compressor positions in Figure 3 of Chapter 7.

FIGURE 6.08A
CENTRIFUGAL COMPRESSOR DRIVEN BY A 2-STAGE GEARBOX

For example, in the case of the gearbox in Figure 6.08A, a good frequency range for gearbox input shaft positions #3 and #4 would be a frequency span of about 60X the 1180 RPM, or a maximum frequency (F_{MAX}) up to approximately 72,000 CPM. Comparing this to the required high frequency gear wear frequency span of 200,000 CPM shown in Figure 6.08A for evaluation up to 3.25X GMF_1, a second measurement (Positions 3HI and 4HI) should also be specified, at least in the horizontal and axial directions up to this high F_{MAX}.

Note that helical gears (like those shown in Figure 6.08A) and bevel gears tend to cause much vibration in the axial direction when wear develops; while worn spur gears normally cause greatest vibration in radial directions (horizontal or vertical).

In addition, the two different frequency spans at each of the gearbox positions may require two separate transducers, and even two separate units of measure (velocity and acceleration). Looking at Figure 6.08A, note that Positions 6HI and 8HI will require analysis up to 540,000 CPM (9000 Hz); whereas the 2950 RPM of this shaft would require analysis only up to about 150,000 CPM (for detection of unbalance, misalignment, looseness, etc.). Therefore, the 150,000 CPM could be accurately evaluated by a magnet mounted accelerometer measuring velocity; whereas the high frequency acceleration measurements up to 540,000 CPM would likely require either a stud mounted or adhesive mounted accelerometer in order to give repeatable, accurate data up to such a high frequency. This is very common with such multi-stage speed increasers.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

Figure 6.08B pictures a large bullgear which is meshing with each of four pinions mounted on the impeller assembly shafts of centrifugal air compressors. Note that even though the bullgear itself meshes with each of the four pinions, there will only be one gear mesh frequency (GMF) since the one common gear (bullgear) meshes with all four other gears.

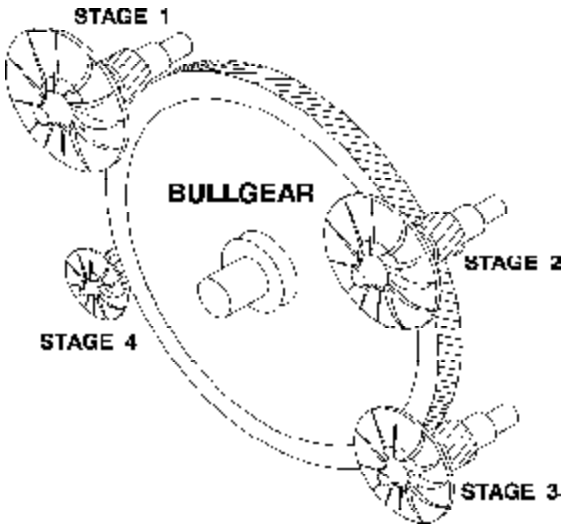


FIGURE 6.08B
PERSPECTIVE VIEW OF AN INGERSOLL-RAND CENTAC CENTRIFUGAL AIR COMPRESSOR BULLGEAR MESHING WITH 4 IMPELLER ASSEMBLY PINIONS (MODEL C-21 2100 ICFM)

COMPONENT	# TEETH	RPM	Hz
Stage 4	25T	49,270	821.13
Stage 3	27T	45,620	760.3
Stage 2	32T	38,490	641.51
Stage 1	42T	29,325	488.75
Bullgear	344T	3,580	59.68

GMF = # Bullgear Teeeth X Bullgear RPM = # Stage 1 Teeth X Stage 1 RPM
GMF = 344T X 3580 RPM = 29,325 RPM X 42T
GMF = 1,231,680 CPM = 20,528 Hz

6.082 Indications of Gear Tooth Wear:

A fairly large number of indications of potential gear problems exist and are summarized in the **Illustrated Vibration Diagnostics Chart** found in the beginning of Chapter 6. Here, we will concentrate on gear tooth wear itself, leaving some of the more exotic analyses to the texts of higher level seminars.

Figure 6.08C will be used to illustrate how gear wear is detected (note that Figure 6.08C pictures a 2-Stage rotary screw compressor including its driving bullgear, timing gears on the low and high pressure stages, and the drive gear arrangement for the gear lubrication pump). Following below will be many of the comments which should be made concerning detection of gear tooth wear:

© Copyright 2001 Technical Associates Of Charlotte, P.C.

1. One of the first indications of gear tooth wear is a pronounced increase in amplitude of the fundamental gear mesh frequency (1X GMF) and/or its harmonics (2X GMF, 3X GMF, etc.). However, one of the problems with watching only the amplitude of gear mesh frequencies is that gear mesh frequencies can sometimes significantly increase or decrease, depending on load (particularly in the case of the fundamental gear mesh frequency which may see an amplitude increase of a factor of 2X or 3X, depending on the machine and gear type).

2. Of even greater concern will be watching for increases in both the amplitude and the number of sideband frequencies surrounding gear mesh frequencies. Such sideband frequencies will be spaced at the RPM of each mating gear. For example, please refer to the plot shown in Figure 6.08D. Here, each of three gear mesh harmonics for the gear pump itself exist and are labeled "OPMF", "2X OPMF" and "3X OPMF". Note particularly the number of sideband frequencies surrounding 2X OPMF. There are four families of sidebands surrounding 2X OPMF with a spacing of approximately the 2719 RPM speed of the pump shaft (F_{OP} in Figure 6.08C).

3. Normally, when gears are in good condition and well aligned with respect to one another, the amplitude of the fundamental gear mesh frequency (1X GMF) will be highest with levels of 2X GMF and 3X GMF falling off. In addition, they will usually only show about one family of sidebands which will be spaced at the RPM of each gear. For example, referring to Figure 6.08C, note the meshing of the 55 tooth gear operating at 1780 RPM and the 36 tooth gear operating at 2719 RPM oil pump shaft speed. In this case, the gear mesh frequency (GMF_{OP}) is 97,900 CPM (55T X 1780 RPM). Therefore, if this mesh is in good shape and well aligned, normally a low amplitude in this case would occur at 97,900 CPM (GMF_{OP}) and its multiples; as well as lower amplitude sideband frequencies spaced in this example at 1780 RPM input speed and at 2719 RPM output speed. Such sidebands would occur on either side of this gear mesh frequency.

   In the case of the spectrum shown in Figure 6.08D, note that each of four sideband families were found to be surrounding 2X oil pump mesh frequency (2X OPMF), each of which were spaced at the oil pump shaft speed of 2719 RPM. Therefore, this indicated a potential problem. In this case, since amplitudes at both the fundamental oil pump mesh frequency (OPMF) and 3X OPMF were low, and since there were so many 2719 RPM sidebands surrounding 2X OPMF, it was concluded that the oil pump gears were possibly misaligned with respect to one another. Subsequent inspection found this was the case as evidenced by wear patterns showing that the load was only being carried by about 25% to 30% of the face width of each gear tooth. Corrective measures were taken to better align these gears, resulting in significantly increased life of the oil pump.

   For comparison purposes, Figure 6.08E was taken on a nearby sister machine (Compressor #1) which showed much lower vibration at 2X OPMF (59,565 CPM) with data taken at the identical location (Position 9A in Figure 6.08C). In the case of Figure 6.11E, note that narrowband envelope alarms were specified for each compressor and were used by the PMP software to detect any alarm violations like that which occurred with the #3 rotary screw compressor with plots shown in Figures 6.08D and 6.08E.

4. Often, initial tooth wear will be detected at 3X gear mesh frequency due to the sliding - rolling - sliding action which takes place during each mesh as described above. Here again, in this case, look not only for significant amplitude increases in 3X GMF, but even greater amplitude increases in sidebands surrounding it; as well as more and more sideband families appearing during subsequent surveys.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

5. If more than one gear in a mesh has worn teeth, sidebands will occur around gear mesh frequencies with a spacing at the RPM of each of the gears having worn teeth. For example, referring back to Figure 6.08C, if both the 32 tooth and 48 tooth timing gears on the high pressure stage become worn, sidebands spaced at both the 32 tooth gear speed of 11,443 RPM, as well as at the 48 tooth gear speed of 7629 RPM would appear around the high pressure gear mesh frequency (GMF_{HP}) of 366,171 CPM and/or its harmonics.

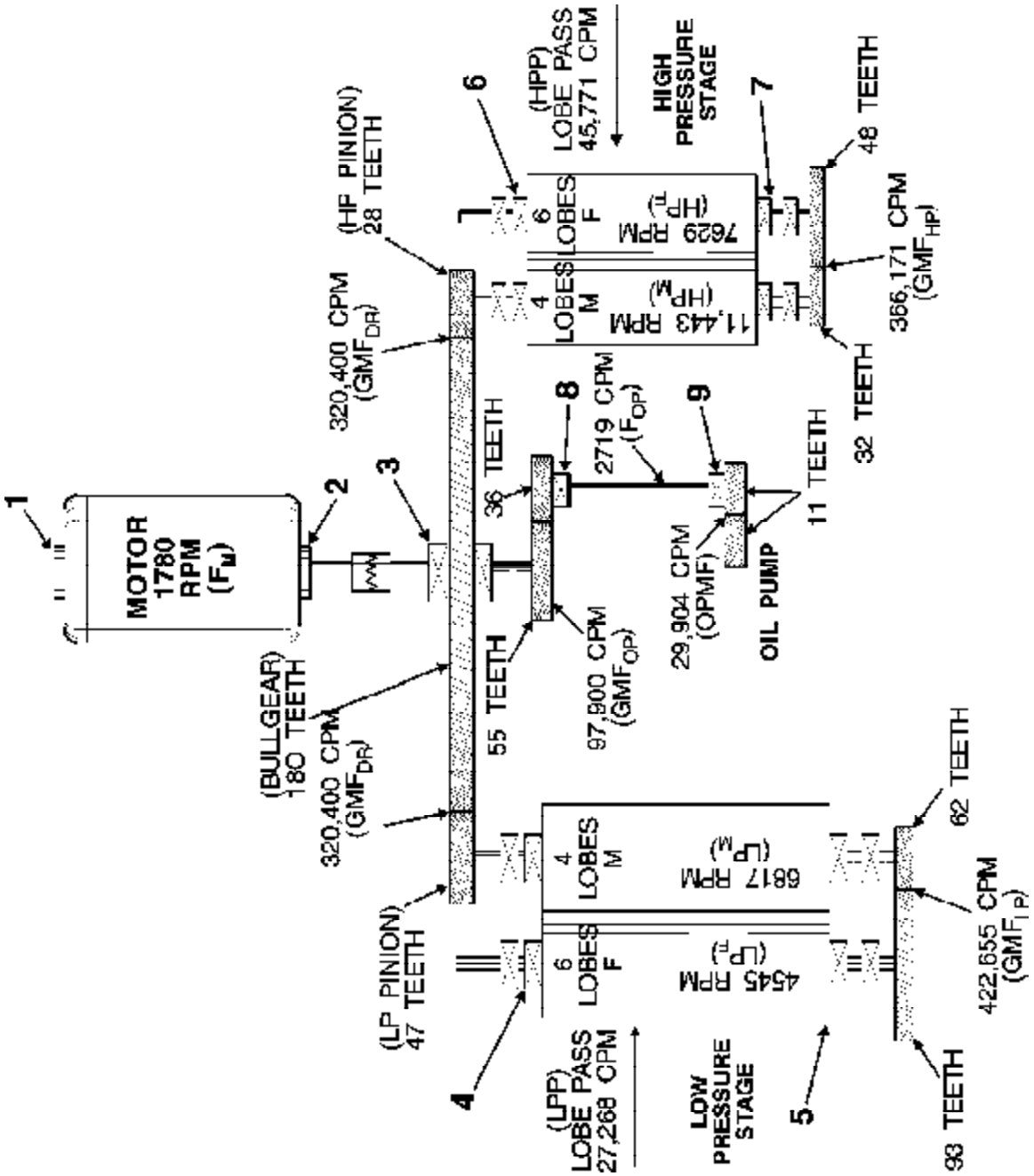


FIGURE 6.08C
ACCELEROMETER LOCATIONS ON A 2-STAGE ROTARY SCREW AIR COMPRESSOR

© Copyright 2001 Technical Associates Of Charlotte, P.C.

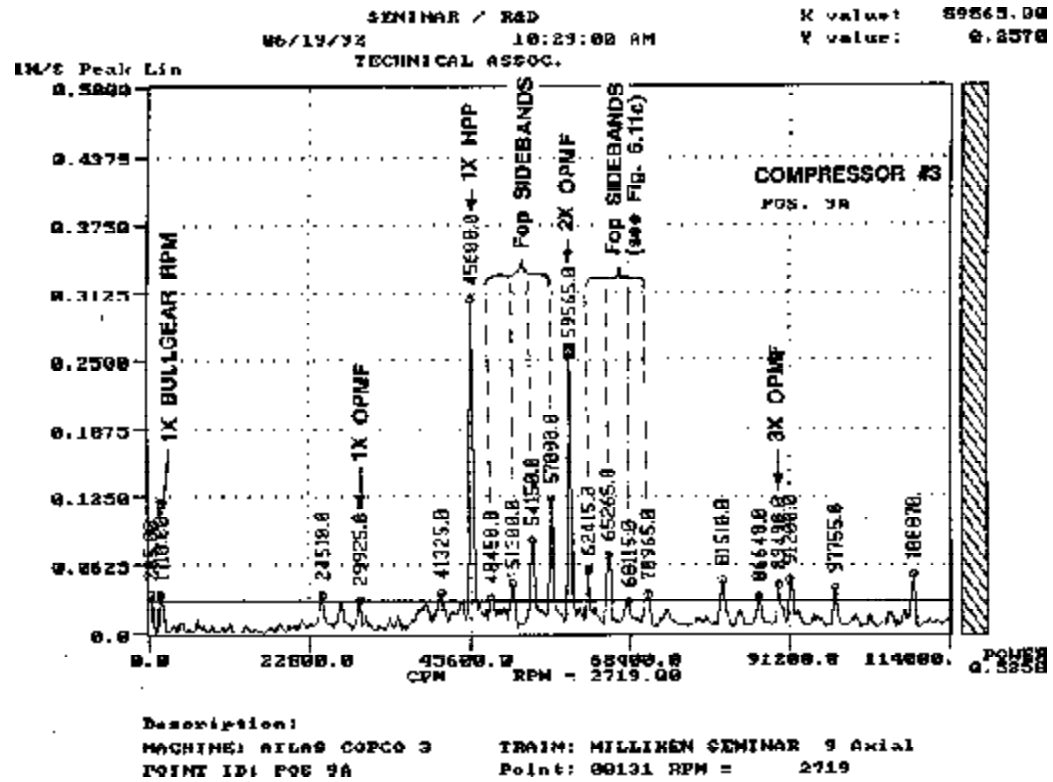


FIGURE 6.08D
THRESHOLD PLOT

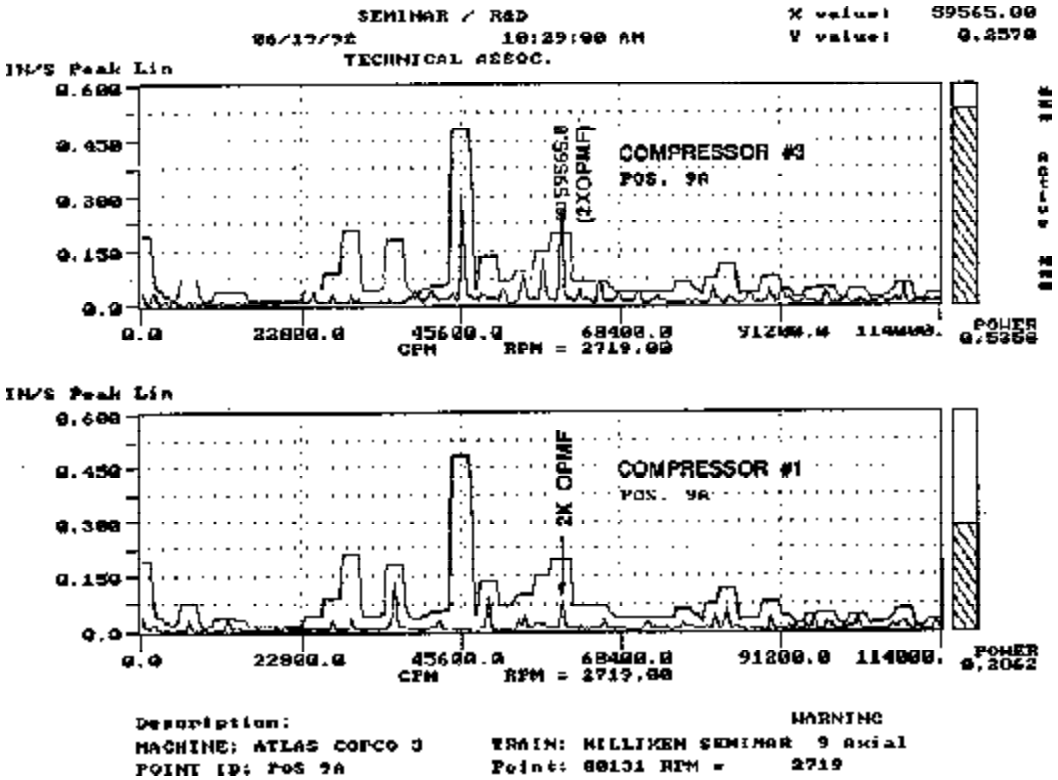


FIGURE 6.08E
NARROWBAND ALARM PLOT COMPARISON

© Copyright 2001 Technical Associates Of Charlotte, P.C.

6.09 INTRODUCTION TO ELECTRICAL PROBLEM DETECTION

Many find it surprising that you can detect not only mechanical problems, but also electrical problems using vibration signature analysis. However, remember that the magnetic fields within a motor create flux which induces electromagnetic forces, and that these, along with forces from mechanically induced problems must all be supported by the bearings. These forces shown in Figure 6.09A can then be measured directly by force transducers placed on the bearing housings, or indirectly by vibration transducers such as accelerometers, velocity pickups or noncontact displacement probes. Accelerometers directly measure the quotient of force divided by mass (f/m), which is acceleration (according to Newton's Law). Again, it does not matter whether these forces themselves are mechanically or electrically induced.

Figures 6.09B and 6.09C picture the stator and rotor of an induction motor, along with a closeup view of a rotor, rotor bars and shorting rings. Note some important formulas that are included in Figure 6.09B. These include the calculation of synchronous speed (N_S) which is dependent on the line frequency (F_L) supplied and the number of poles (P). Also shown is a formula for determining the slip frequency (F_S) which is just the difference in the synchronous speed and actual RPM (for example, in a two pole motor turning 3580 RPM, the slip frequency would be 3600 - 3580, or 20 CPM). Also shown is a formula for calculating what is known as the pole pass frequency (F_P) which equals the slip frequency X the number of poles (in the example given above having 2 poles, the slip frequency would be 20 CPM X 2 poles = 40 CPM). Finally, another formula is given for calculating the rotor bar pass frequency (RBPF) which equals the number of rotor bars X the rotor RPM.

Please refer to Figure 6.09C which shows the three components making up the rotor including the rotor bars, rotor laminations (windings) and shorting rings on either end (also sometimes known as end rings).

The **Illustrated Vibration Diagnostics Chart** given in the beginning of Chapter 6 shows that a large variety of electrical problems can be detected using vibration analysis. In this introductory paper on electrical problem detection, most of the emphasis will be given on how to calculate problems within the stator, rotor problems and variable air gap between the rotor and stator. Note that both stator problems and variable air gap problems are detected by high amplitudes at 2X electrical line frequency ($2F_L$). In fact, probably the majority of electrical problems occur at this same frequency ($2F_L$). The remaining electrical problems included in the **Illustrated Vibration Diagnostics Chart** will be discussed in texts given in higher seminar levels.

6.091 Why many electrical problems occur at 2X line frequency:

Many electrical problems are detected due to "higher than normal" amplitudes at 2X electrical line frequency (also known as synchronous frequency). In the United States, the line frequency is normally set at 60 Hz (3600 CPM), whereas in Europe and other locations, it is at 50 Hz (3000 CPM). Therefore, with respect to the United States, a frequency of great importance when detecting electrical problems will be that at 120 Hz (or 7200 CPM).

Figure 6.09A helps explain why so many electrical problems involve twice line frequency ($2F_L$) rather than the fundamental line frequency. Referring to Figure 6.09A, note that during one rotation of the 3600 CPM stator field of this two-pole motor, the magnetic pull towards the closest pole rises from zero to a maximum 2X per revolution in this rotor which is displaced off center (eccentrically) within the stator. Since the field itself revolves at 3600 revolutions per minute, the magnetic pull reaches a maximum at a frequency of 7200 times per minute (or 7200 CPM). Another way of looking at it is that because the close side of the rotor will first be attracted to the north, and then to the south pole, the force itself will vary at 2X line frequency of the magnetic

© Copyright 2001 Technical Associates Of Charlotte, P.C.

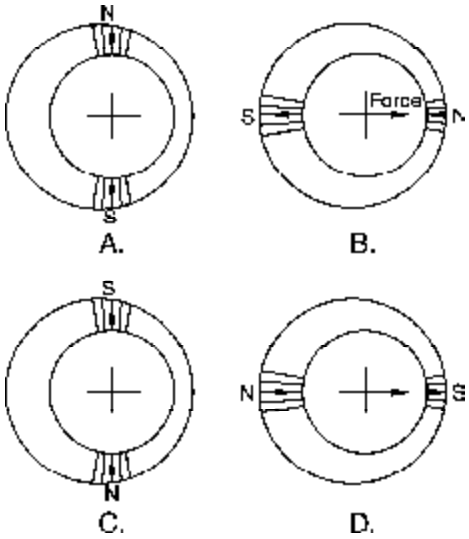


FIGURE 6.09A
UNEVEN AIR GAP BETWEEN MOTOR AND STATOR

field relative to the eccentricity. Therefore, when the rotor is not centered within the stator causing a variable air gap between the rotor and stator, it will always affect 2X line frequency vibration. This fact is true whether you have a two-pole motor or not. That is, if one had a four-pole motor energized with 1800 CPM synchronous line frequency, each of the four poles would pulse as they passed by the small air gap pictured in illustration B of Figure 6.09A generating four pulses per revolution. Then, given the 1800 CPM synchronous frequency of the four-pole motor, this same 7200 CPM frequency will be affected. This same phenomenon would occur with a six-pole motor supplied with 1200 CPM synchronous frequency, still producing 7200 CPM twice line frequency.

Following below will be a discussion on how electrical problems concerning the stator and variable air gap are detected:

6.092 Stator Problems:

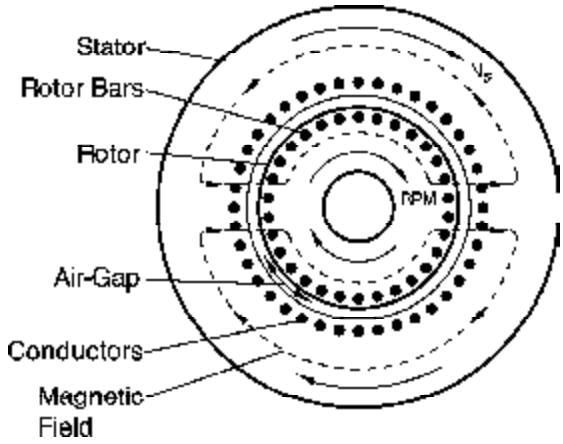
Stator problems detectable by vibration analysis include each of the following:

- Stator Eccentricity (an eccentric stator producing a stationary differential air gap between the rotor and stator)
- Shorted Laminations (insulation problems with lamination layers which can cause localized heating)
- Loose Iron (any looseness or weakness in the stator)

These problems exhibit the following characteristics:

1. All such stator problems generate high vibration at 2X line frequency. However, they do not necessarily generate pole pass frequency sidebands since they originate within the stator, and are not therefore modulated by either running speed or slip frequency. Figures 6.09D and 6.09E indicate a serious electrical problem. Figure 6.12D is a spectrum captured by a data collector during a regularly scheduled PMP route. Note the high amplitude of .230 in/sec at 7200 CPM which exceeded Band 3. A real-time analyzer was then employed on the

© Copyright 2001 Technical Associates Of Charlotte, P.C.



$N_S = \frac{120\ F_L}{P}$ = SYNCHRON. SPEED

$F_S = N_S$ - RPM = SLIP FREQ.

$F_P = (F_S)(P)$ = POLE PASS FREQ.

RBPF = # ROTOR BARS X RPM

WHERE:
F_L = ELECTRICAL LINE FREQ. (often 60 Hz)
RPM = ROTOR SPEED
N_S = SYNCHRONOUS SPEED
F_S = SLIP FREQUENCY (N_S - RPM)
F_P = POLE PASS FREQUENCY
P = # POLES
RBPF = ROTOR BAR PASS FREQUENCY

FIGURE 6.09B
DIAGRAM OF AN INDUCTION MOTOR ALONG WITH ITS STATOR, ROTOR, ROTOR BARS, AIR GAP AND MAGNETIC FIELD

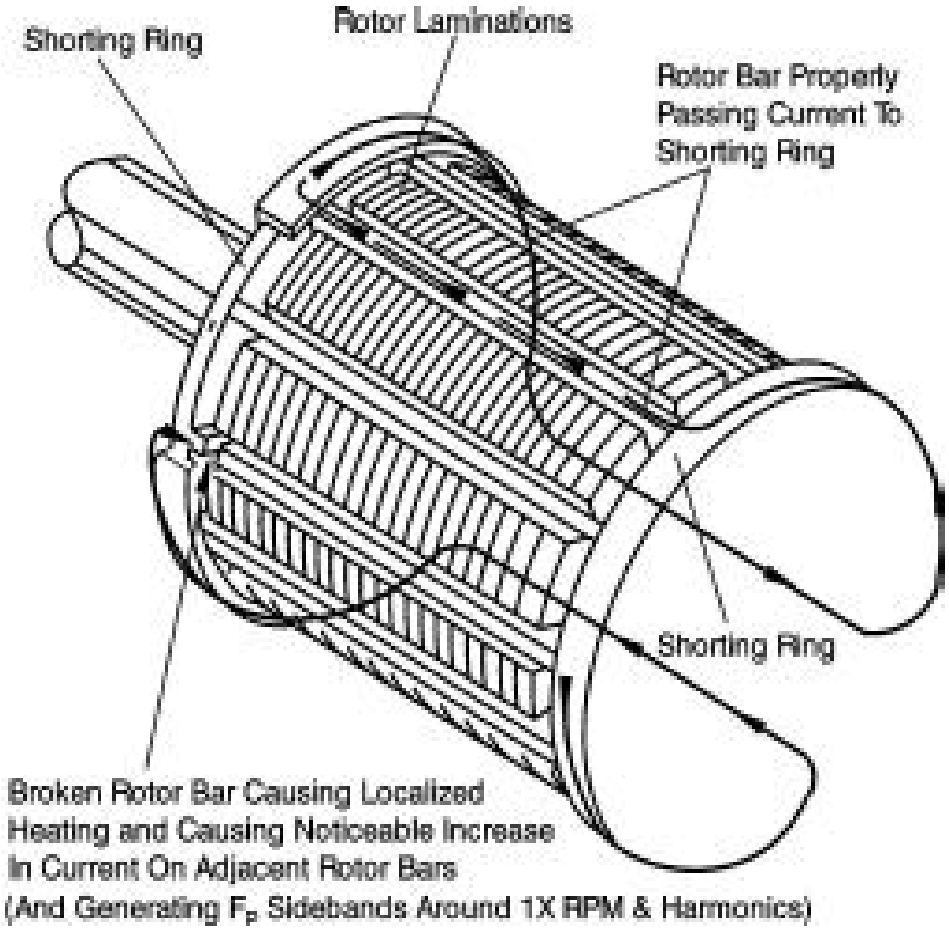


FIGURE 6.09C
ISOMETRIC VIEW OF A ROTOR INCLUDING ITS ROTOR BARS, SHORTING RINGS AND ROTOR LAMINATIONS

© Copyright 2001 Technical Associates Of Charlotte, P.C.

same day to better define this problem which is shown by the zoom spectrum of Figure 6.09E. This spectrum showed a high level of .228 in/sec at 7200 CPM ($2F_L$) and only .0044 in/sec at 2X RPM. Please also note the absence of any pole pass sidebands about 2X line frequency which indicated a stator problem. Upon inspection, a stator eccentricity problem was found.

2. Concern should likely be given motors whose vibration under load exceeds .025 in/sec peak at 2X line frequency on new or rebuilt motors, or .070 in/sec peak on in-service motors having a stator problem (applies to motors in general ranging from 50 HP to 1000 HP). This amplitude applies specifically to the peak at 7200.0 CPM itself ($2F_L$). However, if this motor is directly driving a precision machine tool spindle, $2F_L$ levels must be much lower, on the order of .015 in/sec or less (Note importantly that these amplitudes apply to induction motors running under at least 60 to 70% load).

3. Stator eccentricity produces an uneven stationary air gap between the rotor and stator which results in highly directional vibration, depending on the largest gap differential. The largest magnetic forces occur at a minimum rotor/stator gap. Therefore, the electro-magnetic forces themselves go from a minimum to a maximum each revolution producing vibration at twice line frequency (7200 CPM).

4. Differential air gaps should not exceed 5% for induction motors and 10% for synchronous motors. If the vibration amplitude at $2F_L$ grows over time, the motor should be inspected (if physically possible) by marking a point both on the rotor and the stator. Then, measuring the air gap at the point where the marks align, rotate the rotor in 45° increments and measure the air gap at the point where the stator is marked. If the variation exceeds approximately 5%, the air gap difference is likely due to an eccentric rotor. The next step is to rotate the rotor again in 45° increments, this time measuring the gap at the point where the rotor is marked. If this gap varies more than 5% for these measurements, an eccentric stator is indicated.

5. Loose iron is due to localized stator support weakness or looseness.

6. Shorted stator laminations can cause uneven, localized heating which can actually bow a motor shaft. This produces thermally-induced bow which can significantly grow with operating time, sometimes causing the stator to contact the rotor which can be catastrophic.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

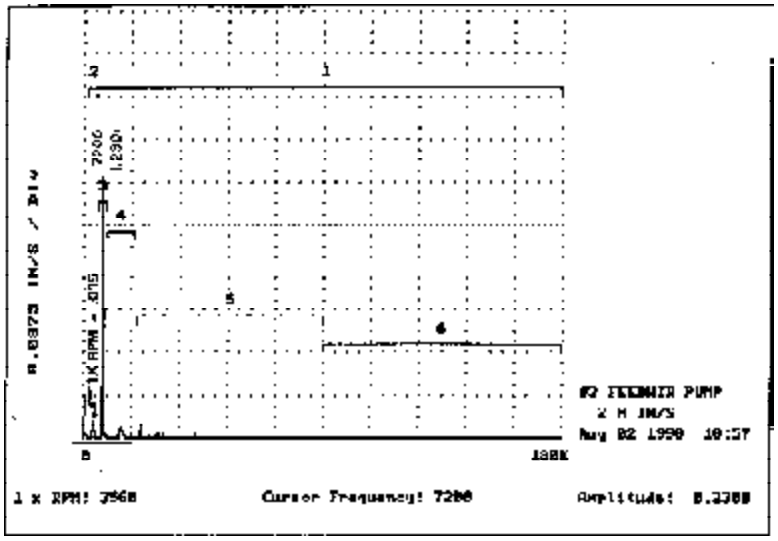


FIGURE 6.09D
NORMAL PMP ROUTE SPECTRUM WITH F_{MAX} = 50 X RPM

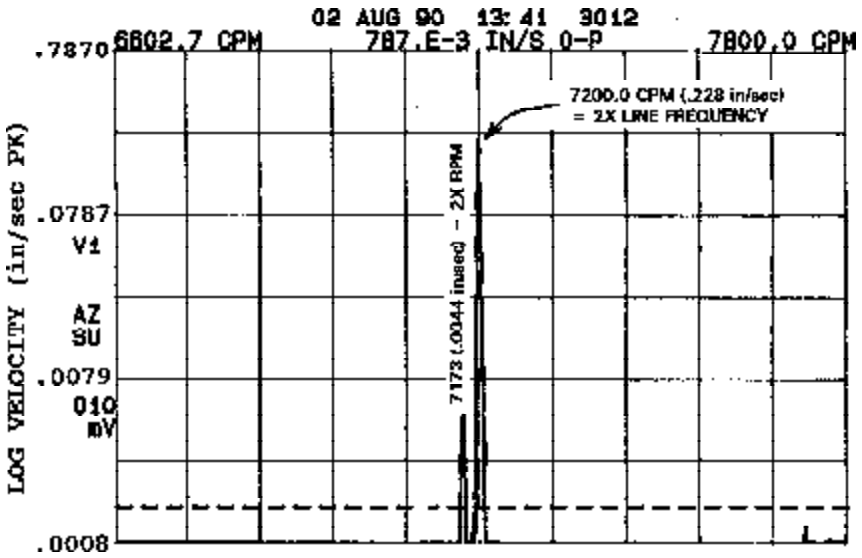


FIGURE 6.09E
ZOOM SPECTRUM INDICATING PROBLEM WITH THE STATOR
(NOTICE THE ABSENCE OF POLE PASS SIDEBANDS)

© Copyright 2001 Technical Associates Of Charlotte, P.C.

6.093 Eccentric Rotor and Variable Air Gap:

In the case of an eccentric rotor, the rotor itself is not concentric with its centerline (see Figure 6.09F). Therefore, an eccentric rotor produces an uneven air gap between the rotor and stator which will rotate with the rotor (as opposed to a stationary air gap in the case of an eccentric stator). An eccentric rotor can be caused by shorted rotor laminations resulting in localized heating and inducing a bowed rotor, or a simple out-of-round rotor.

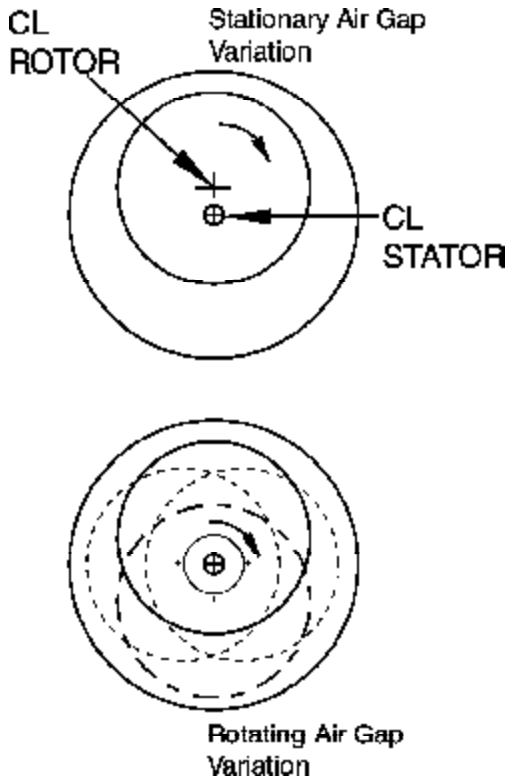


FIGURE 6.09F
STATIONARY & ROTATING AIR GAP VARIATIONS

An eccentric rotor will exhibit the following characteristics:

1. An eccentric rotor most often will produce a high vibration at twice line frequency ($2F_L$) accompanied by sidebands spaced at pole pass frequencies (F_P = # Poles X slip frequency). That is, for a two-pole, 3600 RPM motor, the sidebands will be at twice slip frequency; while for a four-pole motor, sidebands would be at 4X slip frequency. Figures 6.09G and 6.09H show a spectrum indicative of an eccentric rotor problem producing a variable air gap. First, the wideband spectrum of Figure 6.09G showed a high overall of .295 in/sec with .162 in/sec at a "so-called" 7200 CPM frequency. The zoom spectrum of Figure 6.09H revealed a high level of .166 in/sec at 7200 CPM, along with well-formed pole pass sidebands indicative of an eccentric rotor.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

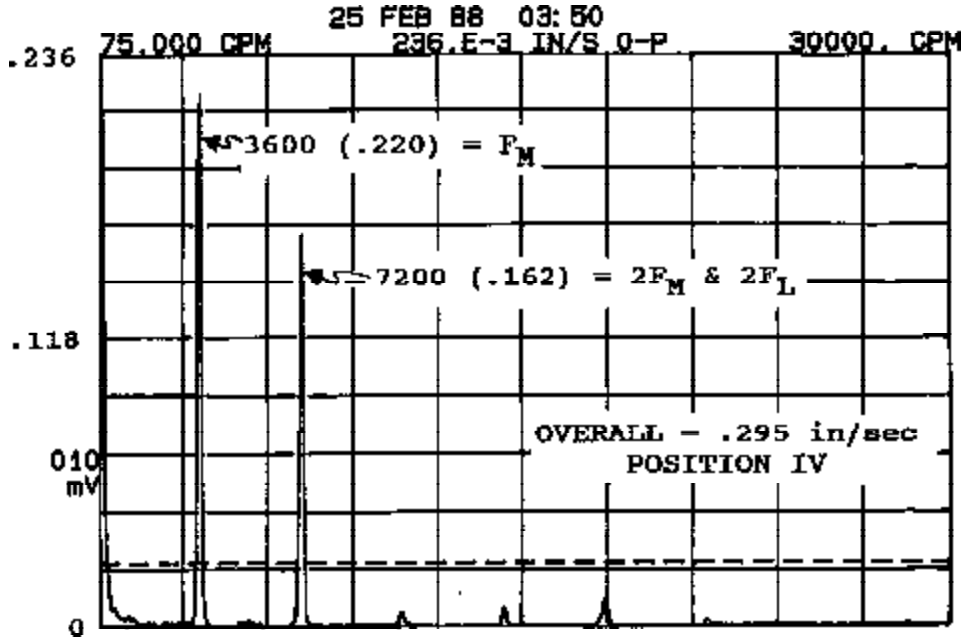


FIGURE 6.09G
30,00 CPM SPECTRUM FOR A CIRCULATING WATER PUMP MOTOR

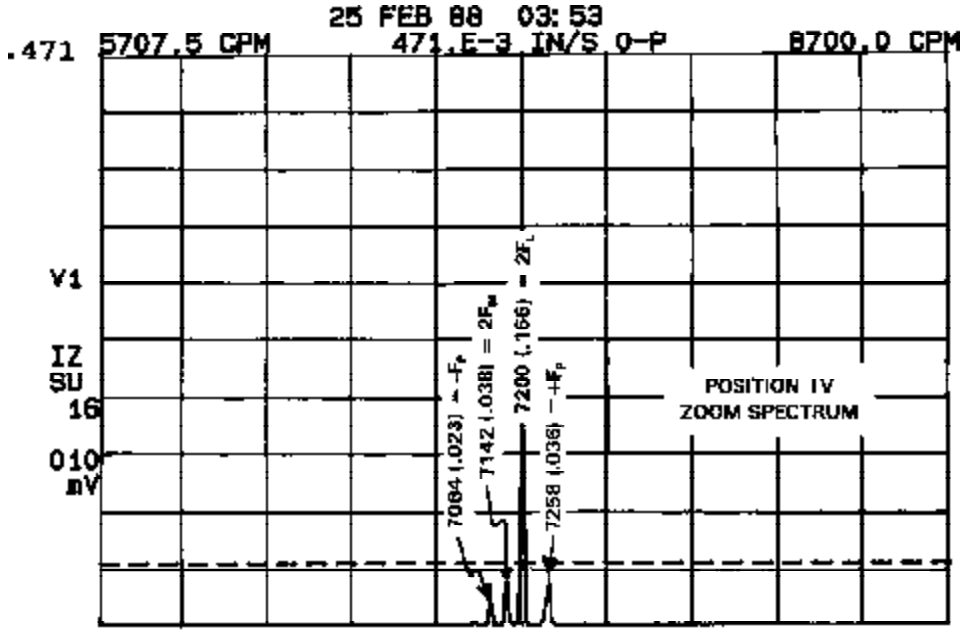


FIGURE 6.09H
ZOOM SPECTRUM INDICATING AN ECCENTRIC ROTOR
(NOT MECHANICAL LOOSENESS OR MISALIGNMENT)

© Copyright 2001 Technical Associates Of Charlotte, P.C.

2. Concern should likely be given motors running under load whose amplitude at twice line frequency (7200 CPM) exceeds approximately .025 in/sec for new or rebuilt motors; or .070 in/sec peak for in-service motors assuming these are general purpose motors. If serving a precision machine tool spindle, the allowable at $2F_L$ would be only approximately .015 in/sec. This would apply to most induction motors ranging in size from approximately 50 HP to 1000 HP. It is important to clarify that this amplitude applies specifically to 2X line frequency itself. Note that these amplitudes apply to induction motors running under at least 60 to 70% load.

3. An eccentric rotor may require adjustment of bearing housings themselves and/or machining of the rotor journals in order to provide an air gap within tolerance all around the periphery. Importantly, an eccentric rotor can also be produced by mechanical problems such as severe misalignment.

4. In a predictive maintenance program, when an eccentric rotor is indicated by pole pass frequency sidebands around $2F_L$, they should be closely trended in future surveys. For example, when a peak at $2F_L$ exceeds roughly .070 in/sec ("Alarm 1"), it should be closely trended in future surveys. If its amplitude noticeably increases above roughly .100 in/sec ("Alarm 2"), and if pole pass sideband amplitudes likewise increase, much greater concern should be given, particularly if even more pole pass frequency sidebands appear above and below $2F_L$ (normally 7200 CPM). On the other hand, if the amplitudes at 7200 CPM and sidebands remain stable over several surveys, no further damage is likely being done to the motor - even if these amplitudes are as much as .175 in/sec at 7200 CPM. In these cases, it will likely be satisfactory just to continue trending. However, the motor itself will likely have a lowered life expectancy.

5. Note that a motor having an eccentric rotor will often experience higher and higher vibration as it comes up to temperature. For example, when such a motor is first started, it may have a level of only .10 in/sec. Then, after about 10 minutes operating time, the amplitude might increase to, for example, .14 in/sec. Next, after about 20 minutes, it may continue to grow up to about .18 in/sec. Finally, after 30 minutes, it may experience possibly .25 in/sec or more. This can be caused by non-uniform heating of the rotor on one side relative to the other. Left uncorrected, it can result in catastrophic failure if the rotor bows sufficiently to cause the rotor to contact the stator.

6.094 Rotor Problems:

Rotor problems detectable by vibration analysis include each of the following:

- Broken/Cracked Rotor Bars or Shorting Rings (see Figure 6.09l)
- Bad High Resistance Joints between Rotor Bars & Shorting Rings
- Shorted Rotor Laminations
- Loose/Open Rotor Bars not making good contact with End Rings

These problems exhibit the following characteristics:

1. Probably the key area of concern for broken or cracked rotor bars, is the presence of pole pass frequency sidebands around 1X RPM and running speed harmonics (in the case of 2-pole motors, these sidebands will be at 2X slip frequency while at 4X slip frequency for 4-pole motors). Figure 6.09l shows spectra typical for a 2-pole motor having serious rotor

© Copyright 2001 Technical Associates Of Charlotte, P.C.

problems. Initially, the 30,000 CPM wideband spectrum gave little hint of a serious problem, seemingly showing mechanical looseness as evidenced by several running speed harmonics (Plot A of Figure 6.09I). However, upon closer examination, the zoom spectra of Plots B thru D of the figure revealed a series of well-formed pole pass sidebands around 1X, 2X and 3X RPM, respectively. These spectra all indicated cracked or broken rotor bars, shorting ring problems or shorted rotor laminations.

2. As suggested by Figure 6.09I, in addition to the pole pass frequency sidebands around 1X RPM, broken or cracked rotor bars and/or high resistance joints can produce pole pass sidebands around higher running speed harmonics up to and including the 2nd, 3rd, 4th and 5th running speed harmonics. In this case, more than 1 rotor bar is often found cracked or broken since there is more than 1 pulse event per revolution. In fact, when zoom spectra were made around 4X RPM up thru 8X RPM in this figure, they also showed multiple pole pass (F_P) sidebands indicating serious rotor problems. Subsequent checks found this motor to have 4 cracked rotor bars and scoring of each shorting ring (likely due to excessive heat).

3. The key area of concern for loose/open rotor bars is vibration at much higher frequencies at rotor bar pass frequency (RBPF) and also harmonics of this frequency:

$$RBPF = \#\ \text{Rotor Bars} \times RPM$$

4. Here, the concern are amplitudes exceeding approximately .06 in/sec at either rotor bar pass frequency (RBPF) or approximately .03 in/sec at 2X RBPF assuming motors are under at least 60 to 70% load. In addition, the sideband spacing around RBPF and its harmonics will be at exactly twice line frequency (2 F_L). The reader is cautioned that while RBPF itself may be acceptable, if he extends the frequency range to encompass 2X RBPF, he may find amplitudes 10 or more times higher than those at RBPF (particularly if arching is occurring between rotor bars and end rings).

For example, Figure 6.109J shows a spectrum for a motor confirmed to have two or more open rotor bars. This motor had 57 rotor bars and operated at a speed of 1793 RPM, giving the fundamental RBPF at about 102,200 CPM. The wide band spectrum of Figure 6.09J showed an amplitude of only .008 in/sec at RBPF. However, the story out at 2X RBPF was completely different. The zoom spectrum of Figure 6.09K showed an excessive .340 in/sec at 204,380 CPM, or 2X RBPF (over 28 times higher amplitude than at 1X RBPF). Importantly, if a maximum frequency high enough only to capture the fundamental RBPF was taken, this problem would have been missed entirely. Again, the key indicators were the excessively high level at 2X RBPF, which was accompanied by sidebands at exactly 7200 CPM ($2F_L$).

5. Sometimes the amplitudes of RBPF or its harmonic will not be the highest amplitude. Instead, one of the sidebands spaced at difference frequencies of 2X line frequency (usually 7200 CPM with 60 Hz F_L) might be highest. This "array" of frequencies will still include RBPF and exactly $2F_L$ sidebands, and will still indicate loose or open rotor bars (assuming 60 Hz F_L) and/or variable air gap.

6. Even though the amplitude of concern in the area of 1X rotor bar pass frequency is about .06 in/sec for most motors, the important task will be first to detect these problems, and then to trend them before making any rash decisions about overhauling motors. If trending of several sets of spectra do not show any real increases, substantial damage is likely not continuing even with RBPF levels of approximately .10 to .15 in/sec. On the other hand, if the rate of change shows substantial increases from one survey to the next, it does indicate rapidly deteriorating condition which does require corrective action.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

7. In Condition Monitoring programs, it is a must to specify each of the following two special points on a route for each motor to detect electrical problems:

   a. Low Frequency Motor Electrical Point:

      F_{MAX} = 12,000 CPM; 3200 FFT lines; 2 Ave and 50% Overlap Processing. This will allow one to separate the true amplitude at $2F_L$ and at motor running speed harmonics. It will also allow one to see if pole pass frequencies (F_P) surround 1X RPM or its harmonics, which could signal significant rotor bar problems (a 400 line spectrum, even with a frequency span of only 12,000 CPM will almost never even show these pole pass frequencies; instead it will simply show only one peak at these running speed harmonics instead of the 3 or more peaks which are truly present at what "appeared" to be only one peak at each running speed harmonic when only a 400 line spectrum was taken.

   b. High Frequency Rotor Bar Pass Evaluation Point:

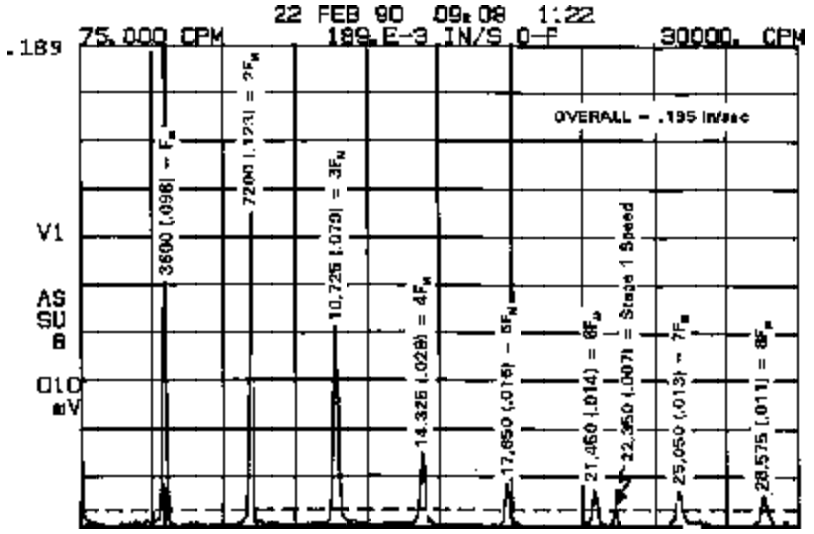
      F_{MAX} = 360,000 CPM; 1600 FFT lines; 8 Ave. (for motors with more than two poles, an F_{MAX} of 240,000 CPM will likely be sufficient to pick up both 1X and 2X RBPF.) This will allow the detection of potential problems at RBPF and its multiples. Look for difference frequencies spaced at exactly $2F_L$ (usually 7200 CPM), even if the number of rotor bars is unknown. Figure 6.12L is a good example. Note the high amplitude peak at RBPF which penetrated both "Alarm 1) (.060 in/sec) and "Alarm 2" (.100 in/sec) narrowband envelopes. Also, notice the 7200 CPM ($2F_L$) sidebands surrounding RBPF. The reader is cautioned to carefully mount his transducer and to employ one with sufficient frequency response out to 360,000 CPM (6000 Hz) if he must make these higher frequency measurements on 2-pole motors.

6.095 Important Closing Comments on Electrical Measurements:

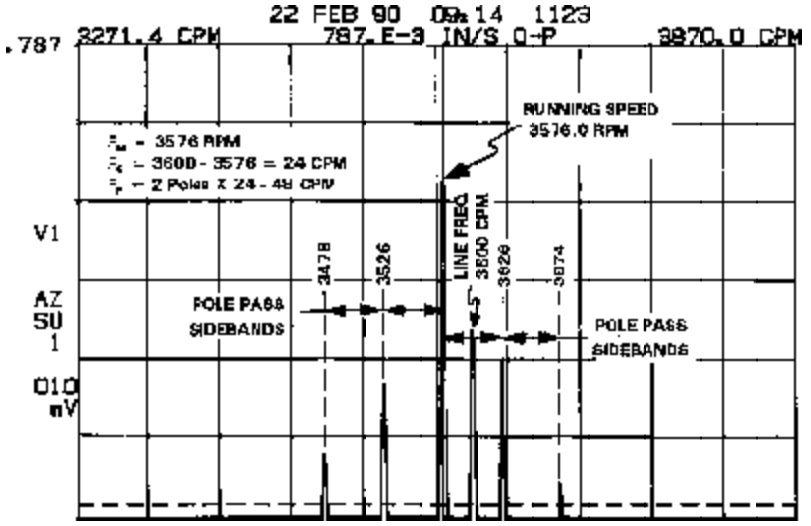
Experience has shown that these high resolution measurements are normally best taken in the horizontal direction. It is important that these measurements be taken with the motor under load (since many of the electrical problems will not even appear if operated unloaded). Remember that these two spectral measurements are in addition to the standard condition monitoring route points on each motor. Experience has shown that taking these two additional points on each motor during a route will require approximately 34 more seconds per motor. There is a cost associated with this additional survey time. However, if one detects only one motor with cracked or broken rotor bars, and prevents catastrophic failure, he can easily save many times this amount not only due to a much lower repair bill, but also due to probable far less down-time. Again experience has shown one very likely will detect such problems if (and possibly only if) he adds these high resolution points to his route. He cannot depend on "catching them with his usual 400 line spectra and THEN going out to perform them. The plain fact is HE WILL NOT EVEN LIKELY DETECT THESE PROBLEMS WITHOUT THEM FOR HE WILL NOT EVEN "SEE" THE POLE PASS FREQUENCY SIDEBANDS!

© Copyright 2001 Technical Associates Of Charlotte, P.C.

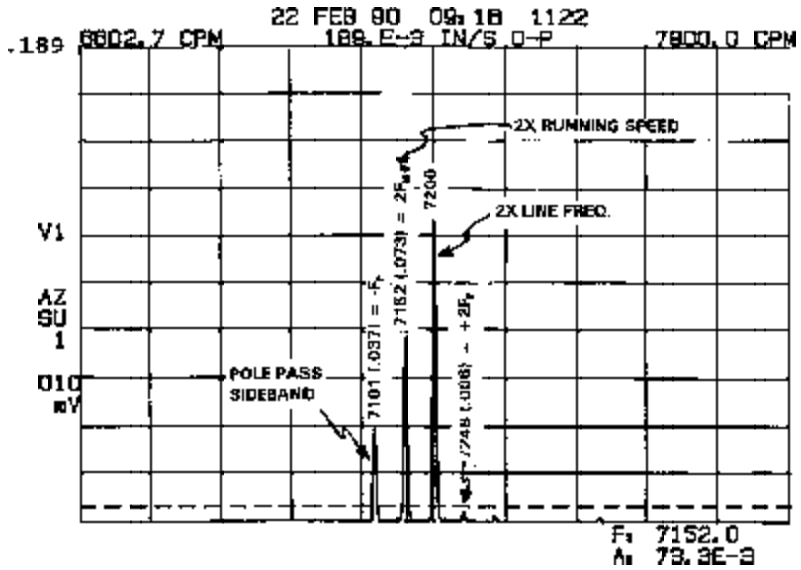
© Copyright 2001 Technical Associates Of Charlotte, P.C.



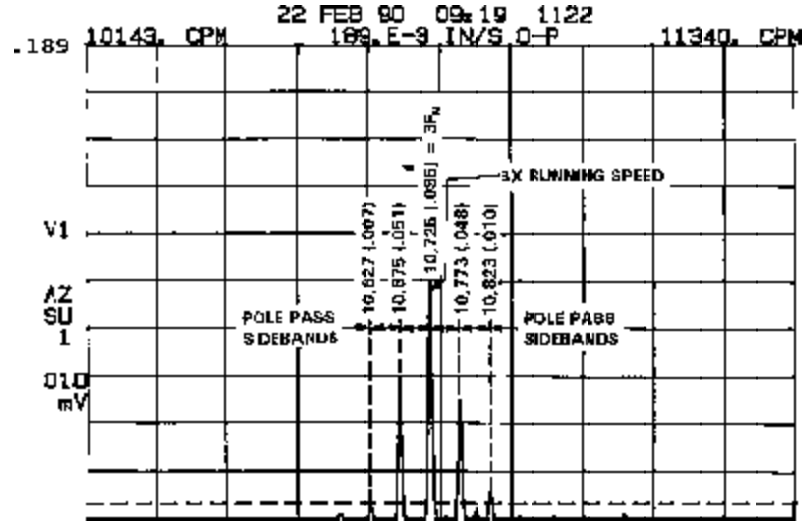
PLOT A
30,000 CPM SPECTRUM ON A CENTAC MOTOR (POS. 2H)



PLOT B
ZOOM SPECTRUM AROUND 1X RPM SHOWING MULTIPLE POLE PASS SIDEBANDS (BROKEN OR CRACKED ROTOR BARS/SHORTING RING PROBLEMS INDICATED)

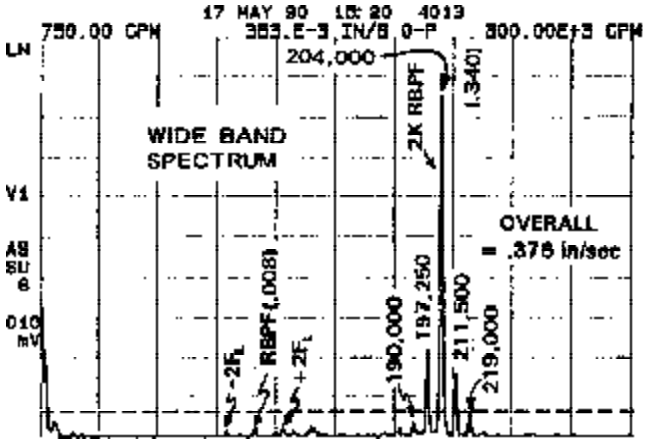


PLOT C
ZOOM SPECTRUM AROUND 2X RPM (POLE PASS SIDEBANDS ALSO HERE)



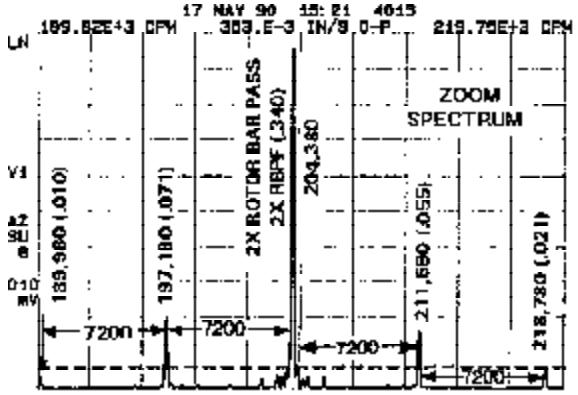
PLOT D
ZOOM SPECTRUM AROUND 3X RPM (NOTE POLE PASS SIDEBANDS HERE ALSO)

FIGURE 6.09I
WIDEBAND & ZOOM SPECTRA FOR A MOTOR HAVING CRACKED OR BROKEN ROTOR BARS OR SHORTING RING PROBLEMS



IN/S 0-P	CPM
77.4E-3	3000.0
77.4E-3	197.24E+3
305.E-3	204.00E+3
56.4E-3	211.49E+3
21.0E-3	219.00E+3

FIGURE 6.09J
SERIOUS ROTOR BAR PROBLEM DETECTED AT 2X RBPF, BUT COMPLETELY MISSED BY 1X ROTOR BAR PASS FREQUENCY

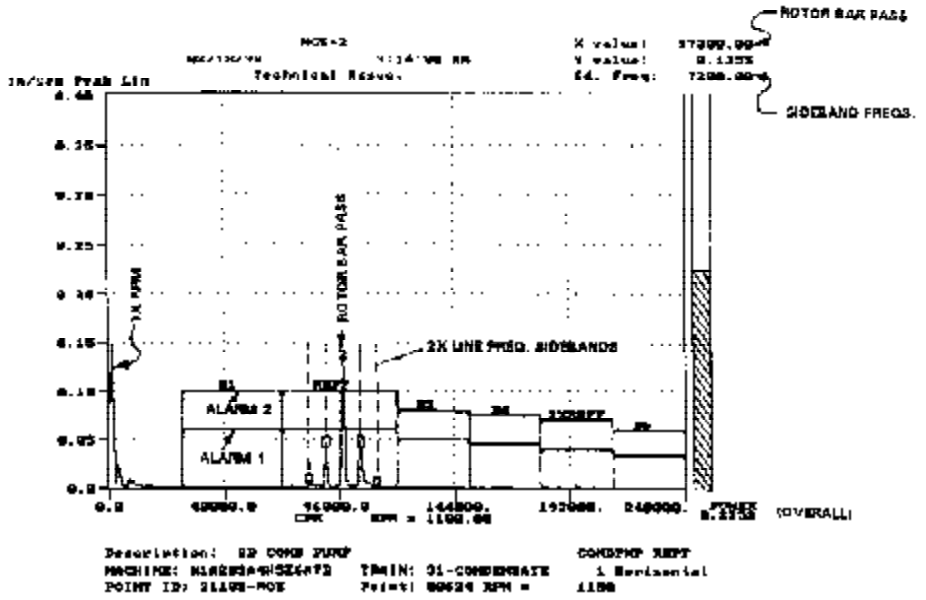


IN/S 0-P	CPM
70.9E-3	197.18E+3
21.0E-3	204.07E+3
340.E-3	204.38E+3
21.0E-3	204.60E+3
54.8E-3	211.58E+3

NOTE: 6-10-90 G.E. Repair Shop phoned to confirm this motor had 2 or more open rotor bars which will likely result in motor replacement.

FIGURE 6.09K
SERIOUS ROTOR BAR PROBLEM DETECTED AT 2X ROTOR BAR PASS FREQUENCY

© Copyright 2001 Technical Associates Of Charlotte, P.C.



02/18/92
9:14:00 AM

Threshold Report

PLANT: NGS-2
Description: 2B COND PUMP
MACHINE: K1A2B3A4H5E6A7B
POINT ID: 31102-MOE
Units: in/sec Peak Lin
Threshold: 0.0050
TRAIN: 31-CONDENSATE
Point: 00624
RPM: 1180.00

Frequency Hz	CPM		Order	Spect Amp	% Exceeded
18.95	1.14E+3		0.96	0.1481	2861.08
61.43	3.69E+3		3.12	0.0237	374.55
138.02	8281		7.02	0.0098	95.67
1383.46	8.30E+4	$-4F_L$	70.35	0.0084	68.81
1503.55	90213	$-2F_L$	76.45	0.0468	835.97
1623.67	97420	= RBPF	82.56	0.1355	2610.03
1743.41	1.05E+5	$+2F_L$	88.65	0.0468	835.97
1863.95	1.12E+5	$+4F_L$	94.78	0.0053	5.24

Power	Sync	Subsync	Non-Sync
0.2253	0.2138	0.0	0.0710

FIGURE 6.09L
TYPICAL SETUP OF 6 SPECTRAL ALARM BANDS FOR A 6-POLE CONDENSATE PUMP MOTOR (1180 RPM NAMEPLATE SPEED)

© Copyright 2001 Technical Associates Of Charlotte, P.C.

6.10 BELT DRIVE PROBLEMS

There are a whole variety of belt-driven machines ranging from those with inherently high vibration such as reciprocating air compressors to those requiring very low levels such as machine tool spindles. However, if the proper precautions are taken, the great majority of such machines should have low vibration. Probably 3 of the greatest factors affecting how much vibration a belt-driven machine will experience are:

- Alignment of Sheaves
- Sheave Concentricity
- Sheave Construction and Attachment Method

If each of these factors are carefully considered, there is no reason to have inherently high vibration in belt-driven machines.

Before discussing belt drive problems detectable by vibration analysis in particular, general statements concerning belt drives should be made:

1. It is best to take radial measurements in line with belt direction as shown in Figure 6.10A.

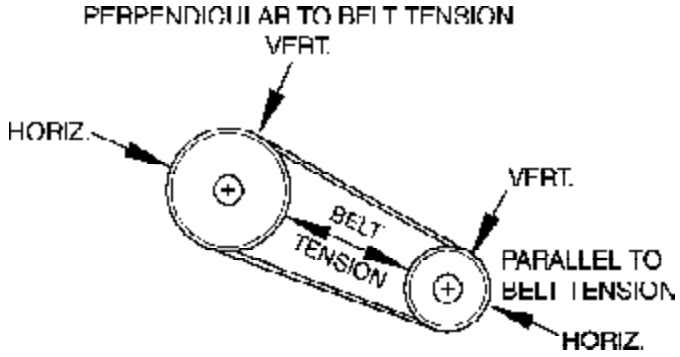


FIGURE 6.10A
PROPER MEASUREMENTS ON A BELT DRIVE

2. Adjustable V-belt sheaves create undue vibration and premature belt and sheave deterioration. These devices have inherent vibration problems since it is not possible to keep sheave faces parallel with one another which allows belts to ride up and down in the grooves with each revolution. As a result, this creates belt tension variation which generates high vibration and accelerates belt and sheave wear.

3. Another critical factor in belt drives is the amount of sheave eccentricity (i.e., runout). Unfortunately, sheaves which are purchased for general utility machinery almost always have inherently high eccentricity much greater than that of other components which are used in general rotating machinery. As a result, as soon as these are assembled to the machine, they themselves can generate high vibration and even cause noticeable variations in belt length and tension with each revolution. It is up to industry to demand tighter concentricity tolerance sheaves and to enforce these specifications right at machine acceptance.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

4. V-belt drives are often blamed as the source of high vibration when, in fact, they are simply reacting to other problems such as unbalance, misalignment, mechanical looseness, etc. When these other problems are present, they can cause high vibration in the belts which themselves are not the source of the problem.

Following below are discussions on many of the belt drive problems which can be detected by vibration analysis:

6.101 Worn, Loose or Mismatched Belts:

A typical spectrum indicating worn belts is given on page 4 of the **Illustrated Vibration Diagnostic Chart**. Note the belt frequency harmonics, all of which happen to be below both the driver and driven RPM in this example. The belt frequency (or belt RPM) is calculated as follows:

$$\text{Belt Freq.} = \frac{(3.142)(\text{Pulley RPM})(\text{Pulley Pitch Dia.})}{\text{Belt Length}}$$

When using the above equation, it is important to enter both the RPM and pitch diameter of the same pulley. It does not matter which pulley is used as long as the variables both come from the same pulley. Note that in all cases the belt RPM will be less than either the driver or driven RPM. However, belt frequency harmonics often will be higher than one or both of these. Worn, loose or mismatched belts display the following characteristics:

1. Worn belt defects detectable by vibration analysis include cracks, broken-off pieces of belt, hard and soft spots, lumps on belt faces, and also a crooked belt which has taken a set deformed shape during packing and storage.

2. When the problem is a worn belt(s), they will normally generate 3 to 4 multiples of belt frequency. Often, the 2X belt frequency peak may be dominant; in other cases, the fundamental belt frequency peak itself may not even show. In addition, worn belts sometimes lift the baseline of the spectrum throughout the subsynchronous frequency region, and just beyond driver and driven speeds as demonstrated in Figure 6.10B. In each case, worn belts normally cause unsteady amplitudes that sometimes pulsate with either the driver or driven RPM if any of these harmonics are close to either the driver or driven speed.

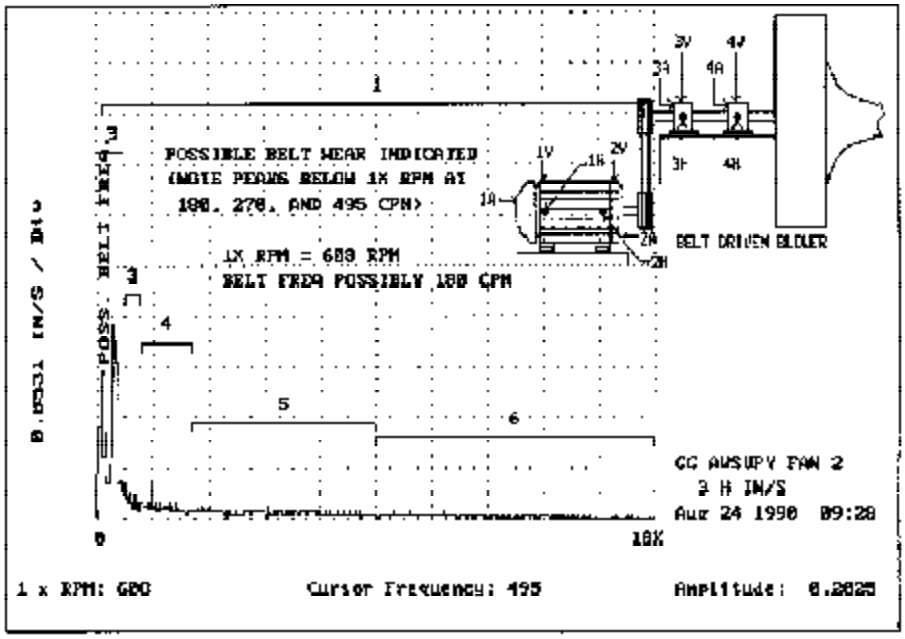
3. Belt defects usually show higher amplitudes in the direction parallel to belt tension. To get an idea how much vibration is caused by the belt defects themselves, compare amplitudes for the belt RPM frequencies themselves in a direction parallel with belt tension versus that in a direction perpendicular to belt tension.

4. Other belt specific problems which show up at belt RPM harmonics include belt width variations which cause the belts to ride up and down pulley grooves, creating vibration due to belt tension variations.

5. A loose cog belt is indicated by high vibration at the #cogs X RPM, and/or high vibration at the cog belt frequency itself which is calculated using the above equation.

6. Multiple V-belt drives can generate high vibration in the axial direction if they are unequally tensioned. This can result in excessive thrust bearing wear. These problems are sometimes solved by replacing several individual belts with one multi-belt which has been molded into one piece. If these are used, they place even greater importance on carefully aligning the sheaves.

© Copyright 2001 Technical Associates Of Charlotte, P.C.



2% x 0.500 = 0.0100 1 x RPM: 608.00

Machine	Pos	Dir	Units	Date
GG AWSUPY FAN 2	3	H	IN/S	Aug 24 1990 09:28

Frequency	1 x RPM	Amplitude	Frequency	1 x RPM	Amplitude
180	0.296	0.1550	2520	4.145	0.0175
270	0.444	0.0900	2610	4.293	0.0100
495	0.814	0.2025	3240	5.329	0.0100
585	0.962	0.1850 ≈ 1X RPM	3375	5.551	0.0125
765	1.258	0.0400	3600	5.921	0.0100
900	1.480	0.0375	5490	9.030	0.0100
1080	1.776	0.0250			
1215	1.998	0.0225			
1305	2.146	0.0150			
1440	2.368	0.0175			
1620	2.664	0.0175			
1800	2.961	0.0400			
1890	3.109	0.0175			
2025	3.331	0.0150			
2160	3.553	0.0125			
2340	3.849	0.0175			
2430	3.997	0.0125			

FIGURE 6.10B

© Copyright 2001 Technical Associates Of Charlotte, P.C.

7. Worn, loose or mismatched belts normally generate highest vibration in the radial direction, particularly in line with belt tension.

6.102 Belt/Sheave Misalignment:

Probably one of the greatest sources of belt drive vibration is misalignment of driver and driven sheaves. It is often amazing to see how much vibration can be reduced simply by employing a chalk line between sheaves. Misaligned sheaves display the following characteristics:

1. Axial vibration is most always generated by sheave misalignment which can highly accelerate the rate of wear of thrust bearings.

2. Misaligned sheaves produce high vibration at 1X RPM, predominately in the axial direction. Dominant vibration is quite often at driver RPM, but occasionally at driven RPM. The ratio of amplitudes of driver to driven RPM depends exactly on where the data is taken as well as on relative mass and frame stiffness.

3. Often with sheave misalignment, the highest axial vibration on the motor will be at fan RPM, while the highest axial when measuring on the fan will be at motor RPM. However, this is not always the case.

6.103 Eccentric Sheaves:

Eccentric sheaves are one of the greatest contributors to high vibration in belt-driven machines today, often due to a lack of emphasis on specifying good concentricity in purchase specs. Sheave eccentricity displays the following characteristics:

1. Eccentric and/or unbalanced sheaves cause high vibration at 1X RPM of the eccentric sheave.

2. Highest amplitude is normally in line with the belts and should show up on both the driver and driven sheaves.

3. Unlike unbalance, reaction forces caused by an eccentric pulley are not equally applied throughout the entire 360° rotation of the pulley. The force instead is concentrated in the direction of belt tension along a line passing through the centerline of the 2 shafts. As a result, this highly directional vibration will show comparative horizontal and vertical phase readings which either are identical or 180° opposite one another, depending on which side of the bearing the transducer is located for the measurement. In any case, both phase readings show that the bearing is moving in one line.

4. Since the forces are so directional in nature, the resultant vibration cannot be totally corrected by balancing via attaching washers to taperlock bolts. Even if balanced, the sheave eccentricity will still induce vibration in the belt due to belt length and tension variations, and will result in premature accelerated wear of belts and/or sheaves along with the driver and driven bearings.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

6.104 Belt Resonance:

Just like everything else in nature, belts themselves have a natural frequency which corresponds to the stiffness of the belt, the belt mass, and how much deflection is induced in the belt during operation. The natural frequency of a belt can be determined simply by pulling on the belt, releasing it and measuring the response. Belt resonance will display the following characteristics:

1. A spectrum indicating a belt resonance problem is shown on page 5 of the Table 6.0 **Illustrated Vibration Diagnostic Chart**. In this example, the belt natural frequency lies very close to 1X RPM of either the driver or driven machine. If this occurs, this will cause great flapping of the belt, particularly on the tension side at a frequency corresponding to the belt natural frequency.

2. Not only can a pulley speed excite a belt resonance, but also this can occur if a belt RPM harmonic should line up with its natural frequency.

3. Both the amplitude and phase of vibration at the belt resonant frequency will be unsteady.

4. The belt natural frequency itself can be changed either by altering the belt tension, belt length, sheave center distance, adding an idler pulley, etc..

6.105 Excessive Motor Vibration At Fan Speed Due to Motor Frame/Foundation Resonance:

Often, in the case of belt-driven machines, there will be excessive vibration on the motor. However, when a spectrum analysis is taken, it will show low vibration on the motor at motor speed, but high vibration on it at fan RPM. This can occur in either radial or axial directions. This will be evident by an excessive vibration at 1X RPM of the driven unit, particularly in one direction on the motor. Often, when this occurs, the problem source is excitation of a motor frame or foundation natural frequency by the incoming fan speed vibration. This can be confirmed by simple impulse natural frequency tests explained in Section 6.05. In this case, the solution will normally be stiffening of the frame or foundation by addition of bracing or addition of concrete on the base if it is resonant.

6.106 Loose Pulley or Fan Hub:

Excessive vibration sometimes occurs in belt-driven machines due to looseness either of a pulley or a fan hub itself. This is evidenced by each of the following characteristics:

1. There will be excessive vibration particularly at 1X RPM, but also at several running speeds harmonics.

2. One of the best indicators will be unstable phase. If operating properly, the phase in horizontal, vertical and axial directions should be steady. If a fan hub held on by set screws is loose on the shaft, it may show a difference in both phase and amplitude each time the unit is started up. In these cases, balancing would be only a temporary solution. It may help for a “couple of hours”, but later when the fan hub or pulley rotates slightly on the shaft, it will upset the whole balance possibly dramatically changing the phase and amplitude. This same thing can occur if either the pulley or fan hub has a taper fit, and is not properly pulled up and fastened to the shaft.

3. The solution to each of these cases is ensuring that all rotors attached to the shaft are securely fastened, and properly oriented in the case of a taper fit.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

REFERENCES

1. Parker, Sybil P., Editor and Chief; McGraw-Hill Dictionary of Mechanical and Design Engineering; McGraw-Hill Book Company; New York, NY; 1984.

2. Mitchell, John S.; An Introduction To Machinery Analysis and Monitoring; Pennwell Publishing Company; Tulsa, Oklahoma; 1981; Pages 141 - 151, 172 - 204.

3. Piotrowski, John D.; "Alignment Condition and Its Effect on The Vibration Response of Rotating Machinery"; Vibrations, Volume 1, No. 4; March, 1986; Pages 11 - 17.

4. Buscarello, Ralph T.; Practical Solutions to Machinery and Maintenance Vibration Problems; Published by Update International, Inc.; Denver, CO; Revised April, 1987.

5. Buehler, Mark W. and Bertin, C. D.; Proceedings Machinery Vibration Monitoring and Analysis Meeting - The Vibration Institute; Houston, TX; April 19 - 21, 1983; Pages 191 - 206.

6. Bently, Donald E.; President and Chairman, Bently Nevada, Inc.; "Rubs Research - Studies Reveal Physical Phenomena of Rotor Rubs"; Orbit, Volume 4, No. 3; October, 1983; Pages 3 - 5.

7. Maxwell, J. Howard; "Induction Motor Magnetic Vibration"; Proceedings Machinery Vibration Monitoring and Analysis Meeting - The Vibration Institute; April, 1983, Pages 39 - 51.

8. Advanced Audio-Visual Customer Training Instruction Manual - "Vibration Measurement and Analysis"; IRD Mechanalysis; Columbus, OH; 1985.

9. Makay, Elemer and Barrett, J. A.; "Field Experience Brings Help to Embattled Pump Users"; Power Magazine; July, 1987; Pages 27 - 30.

10. Salamone, Dana J.; "Introduction to Hydrodynamic Journal Bearings"; Mini-Course Notes Machinery Vibration Monitoring and Analysis Meeting - The Vibration Institute; New Orleans, LA; May 22 - 24, 1985; Pages 41 - 56.

11. Middleton, Ben; "Rolling Element Bearing Failure Detection Methods"; Presented at the Acoustical Society of America, Raleigh, NC, October 8 - 9, 1987; Pages 1 - 14.

12. Taylor, James I.; "Determination of Antifriction Bearing Condition by Spectral Analysis"; Sixth in a Series of Technology Interchange on Machinery Vibration Monitoring and Analysis; The Vibration Institute; Clarendon Hills, IL; Pages 1 - 26; 1978.

13. Szrom, David B.; "Low Speed Bearing Analysis"; Proceedings Tenth Annual Meeting - The Vibration Institute; Las Vegas, NE; June 24 - 26, 1986; Pages 183 - 188.

14. Springer, C. W.; "Spectral Analysis of Double-Row Antifriction Bearings"; Vibrations; Volume 4, No. 1; March, 1988; Pages 16 - 17.

15. Springer, C. W.; "The Role of the Time Domain in Analyzing Bearing Defects"; Vibrations; Volume 4, No. 3; September, 1988; Pages 14 - 15.

16. Berggren, J. Charles; "Diagnosing Faults in Rolling Element Bearings - Part I. Assessing Bearing Condition"; Vibrations; Volume 4, No. 1; March, 1988; Pages 5 - 14.
And "Diagnosing Faults in Rolling Element Bearings - Part II. Alternative Analytical Methods"; Vibrations; Volume 4, No. 2; June, 1988; Pages 12 - 23.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

17. Bently, Donald E.; Zimmer, Siegfried; Palmatier, George E.; and Muszynska, Agnes; “Interpreting Vibration Information from Rotating Machinery”; Sound and Vibration Magazine; Volume 20, No. 2; February, 1986; Pages 14 - 23.

18. Schlitz, Richard L.; “Forcing Frequency Identification of Rolling Element Bearings”; Sound and Vibration Magazine; Volume 24, No. 5; May, 1990.

19. Berggren, J. Charles; “Diagnosing Faults in Rolling Element Bearings, Part III. Electronic Data Collector Applications”; Vibrations; Volume 5, No. 2; June, 1989; Pages 8 - 19.

20. Fundamentals of Balancing; Published by Schenck Trebel Corp; Deer Park, L.I., New York; Second Edition; March, 1983, Pages 46 - 47.

21. ”Vibration and Noise Analysis, Dynamic Balancing and Preventive Maintenance”; Audio-Visual Customer Training Instruction Manual; Published by IRD Mechanalysis, Inc.; Columbus, OH; 1975; Page 108.

22. Fasig, Paul; Gahagan, Richard; and Abernathy, Joe; “DC Motor and SCR Firing Vibrations” (Article Submitted to Vibration Institute); 1989; Pages 1 - 12.

23. Szrom, David B.; “Determining Gear Condition With FFT Spectrum Analysis”; Proceedings 11th Annual Meeting - The Vibration Institute; June, 1987; Pages 1 - 5.

24. Winterton, John G.; “Component Identification of Gear Generated Spectra”; Proceedings 11th Annual Meeting - The Vibration Institute; June, 1987; Pages 11 - 17.

25. Vibration Technology - 1; Published by IRD Mechanalysis; Columbus, OH; 1989; Pages 2-2 thru 2-10; 3-2 thru 3-15; and 5-2 thru 5-31.

26. Berry, James E., P.E.; Technical Associates of Charlotte, P.C.; Charlotte, NC; "Tracking of Rolling Element Bearing Failure Stages Using Vibration and High Frequency Enveloping and Demodulated Spectral Techniques"; Analysis II - Concentrated Vibration Signature Analysis and Related Condition Monitoring Techniques Seminar Text; Pages 6-78 thru 6-137; 1993.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

CHAPTER 7

PROVEN METHOD FOR SPECIFYING BOTH SPECTRAL ALARM BANDS AS WELL AS NARROWBAND ALARM ENVELOPES USING TODAY'S CONDITION MONITORING SOFTWARE SYSTEMS

By: James E. Berry, P.E.
Vice-President
Technical Associates of Charlotte, P.C.
Charlotte, NC

James E. Berry has 25 years mechanical engineering experience including 23 years specializing in Machinery Vibration Diagnostics, setup and implementation of Condition Monitoring Programs, Modal Analysis and Stress Analysis. Mr. Berry received both his Bachelor of Science (1973) and Masters Degrees (1974) in Mechanical Engineering from North Carolina State University. He is also a registered professional engineer and is an active member of the Vibration Institute. He has published several vibration analysis articles in technical journals such as Sound and Vibration Magazine, given technical papers and seminars to the Vibration Institute, and has given presentations to several engineering societies including ASME, AIPE, ASA, and ASTME. Serving as a Consulting Engineer, he has performed vibration analysis on a wide variety of both process and utility machinery for a diverse group of clientele served by Technical Associates. He has also authored four seminar texts focusing on vibration analysis and condition monitoring and has been providing professional training services for 12 years.

7.0 ABSTRACT

The fourth edition of this paper has been written primarily with the objective of not only expanding the coverage of Spectral Band Alarm setups to encompass additional machine types, but also to refine those previously established for machine types covered before. Comprehensive statistical analyses have been conducted to help the user specify meaningful overall alarm levels as well as spectral band alarm levels. Here again, this paper is intended to give the analyst a good starting point; such spectral bands and alarm levels should be reviewed after a sufficient quantity of surveys have been conducted, modifying those which are found to need refinement.

Although there is much literature available today on how to diagnose machine problems using vibration analysis, there is little material available on how to specify effective spectral alarm bands on various types of machinery. These spectral alarm bands are now offered within the software of several vendors serving the field of condition monitoring, and thanks to these vendors, provide the potential of detecting numerous machine problems that might otherwise go unnoticed. In the detection and analysis process, if these spectral bands are utilized, they can save the user thousands of dollars in maintenance expenditures and make significant impact on improving plant profitability. First, one needs to know that his machine has a problem. Then, he must take steps to diagnose both the source of the problem and determine its severity. The purpose of this paper is to provide a documented technique on how to specify peak velocity spectral alarm levels and frequency bands for measurements taken on the housings of general process and utility machinery. If properly specified for the specific machine type, drive configuration, bearing type and operating speed, these spectral

© Copyright 2001 Technical Associates Of Charlotte, P.C.

alarm bands will notify the user that he has a problem without generating a series of false alarm "emergencies". The techniques included are not intended, and will not apply to all machine types under all operating conditions. However, they have been successfully applied to a diverse array of machinery ranging from common pumps and blowers to refrigeration chillers, hammer mills, machine tool spindles, high-speed centrifugal air compressors, moderate speed rolling mill drives, etc.

This paper now applies not only to those software systems which allow a number of spectral alarm bands ("power bands"), but also to those which allow generation of narrowband spectral alarms which place a threshold envelope around individual frequencies which are caused by specific sources within the machine ("threshold band alarms").

The complete tabulated procedure for properly specifying spectral alarm bands is included in Table III. It is then followed by several examples which include complete specification of spectral alarm bands for various machine types.

7.1 INTRODUCTION TO SPECIFYING SPECTRAL ALARM BANDS & FREQUENCY RANGES

Properly specified spectral alarm bands are probably one of the most critical weapons in the condition monitoring arsenal today for detecting potentially serious problems which develop in machinery. However, although thousands of data collectors and software are now in place throughout the world having the capability of comparing each new FFT spectrum captured to these user-defined spectral alarm bands, surveys have shown that very few users have sufficient experience to know how to properly and effectively set up these bands in their computers. In fact, a large percentage of plants do not use spectral alarm band capabilities even though they are offered in their software. Instead, they depend on trending of overall levels to warn of impending machine problems. On the other hand, many of those who themselves have made concerted efforts to specify and use these bands often complain that they do not feel very comfortable with the bands they have specified; and do not have the time required to learn how to specify one set for one machine type and an entirely different set for another, depending on how the machine is configured (bearing type, operating speed, drive configuration, etc.). Many users at these plants are hard pressed just to determine what overall alarm vibration levels should be specified for these machines, much less have the time to research how to specify individual band frequency ranges and alarm levels. Many spend hundreds of man-hours just trying to specify the optimum overall, and even then, doubt how meaningful these are.

As we now know from spectrum analysis, if it were possible to obtain so-called "perfect overall vibration specifications", potentially serious problems can still develop within machines, and yet, can have negligible effect on the level of overall vibration. However, these same problems would be noticeable in the FFT spectrum. But, if no spectral alarm system were in place to even detect the presence of a problem, the user would be unaware of its existence until possibly considerable damage had been done, not only to one component, but possibly to several other components in this machine as well.

Such problems as deteriorating bearings, gears, electrical problems, etc. may not make themselves known for some time to those who depend only on overall levels to detect problems. For example, a bearing defect frequency might grow by a factor of 4X from .03 to .12 in/sec and cause almost no change in the overall if amplitudes at 1X and 2X RPM were, say .35 in/sec and .20 in/sec (in this case, the 4X increase by the bearing frequency would only increase the overall from approximately .40 to .42 in/sec). Even though there would be a definite increase in bearing defect vibration, the overall alone would simply not be sufficiently sensitive to show there was any real change.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

Therefore, the express purpose of this paper is to offer a procedure for specifying meaningful spectral alarm bands for a variety of machine types and configurations. This method has been in use by our company for numerous years as we have set up and carried out complete condition monitoring programs for a large number of clients in a diverse array of industries. This technique is summarized in Table III.

This technique is not meant (or implied) to specify concrete spectral alarm bands that cannot be altered, no matter what the unique spectral characteristics of a given machine family. Instead, it is meant to provide the analyst with a firm starting point to:

(1) Allow a plant having no prior experience or machine vibration history to initially set up effective spectral alarm bands for hundreds of machines in his plant prior to making baseline (or initial) measurements;

(2) Allow a more experienced plant to set up spectral alarm bands for the first time, even though the plant might have captured data on large numbers of machines for several years, but has never set up the bands due to a lack of understanding on how to properly do so;

(3) Allow the plant which in fact has installed spectral alarm bands to objectively compare them to other setups, and to evaluate how effective their current bands are.

Please keep in mind that after several samples of data have been acquired, the user should carefully review how each setup for each grouping of machines is working (assuming it is possible to place all his machines into specific groupings or families). Included in the paper will not only be how to specify spectral alarm bands, but also suggestions on how to evaluate their effectiveness and refine them as well.

7.11 TWO TYPES OF SPECTRAL ALARM BANDS

Importantly, note that two different types of spectral alarm bands are used by several different condition monitoring software versions - 1) Absolute Threshold, and 2) so-called "Power Band" type. It is important that the user know which type is employed by his predictive maintenance software system and take this into account.

Absolute threshold systems enable users to specify the maximum allowable amplitude of any single peak within each band. If any peak equals or exceeds this threshold, it will cause the band to go into alarm.

On the other hand, power band systems calculate the total energy (or "power") within each band generated by all the peaks within that band. The total power within each band is calculated using the same equation as that used to determine the overall level of an entire spectrum as per the following formula:

$$OA = \frac{\sqrt{\sum_{i=1}^{n} A_i^2}}{\sqrt{N_{BF}}} = \frac{\sqrt{A_1^2 + A_2^2 + A_3^2 + ... + A_N^2}}{\sqrt{1.5}}$$

EQUATION (1)

Where:

OA = Overall Level of Spectrum or Band
n = Number of FFT Lines of Resolution
A_1 = Amplitude of Each of the FFT Lines
N_{BF} = Noise Bandwidth for Window Chosen
1.5 for Hanning Window

Which means $OA = .8165 \sqrt{\sum_{i=1}^{n} A_i^2}$

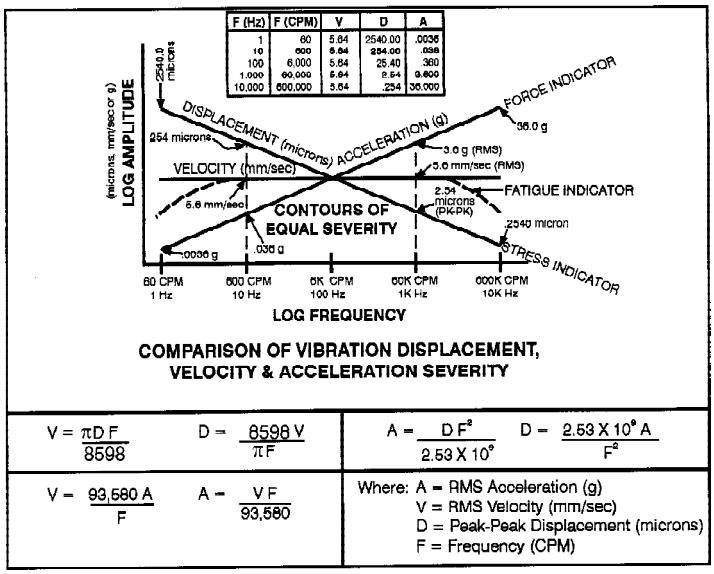
© Copyright 2001 Technical Associates Of Charlotte, P.C.

Note that it is not necessary for any individual peak to equal or exceed a power band alarm in order to exceed the alarm for that band. That is, if a band were specified to extend from .50X F_{MAX} through F_{MAX} using a 400 line FFT, Equation (1) would be used to calculate the power from the 200th line through the 400th line of the spectrum. For example, for a power band having a .20 in/sec specified alarm, the energy of only 2 peaks within this band having amplitudes of .175 and .185 in/sec, respectively, would likely exceed this alarm.

Importantly, the spectral alarm band amplitudes specified in Table III assume the power band type. The same frequency ranges would apply for either alarm band system. However, if one had absolute threshold bands, he should lower alarm levels specified in Table III somewhat - probably on the order of 20% to 30%, particularly for those bands having wider frequency ranges.

7.12 WHICH VIBRATION PARAMETER TO USE IN SPECTRAL ALARM BANDS - DISPLACEMENT, VELOCITY OR ACCELERATION?

Three important items must be understood when setting up spectral alarm bands for machines. First, one must know what forcing frequencies will be generated by such things as rolling element bearing wear, sleeve bearing wear, gear problems, electrical problems, unbalance, misalignment, etc. Secondly, however, he must then know which vibration parameter (displacement, velocity or acceleration) will best detect those problems he will see on his particular machines. Thirdly, he must know how many FFT lines must be used to even show the presence of such problems.



COMPARISON OF VIBRATION DISPLACEMENT, VELOCITY & ACCELERATION SEVERITY

$V = \frac{\pi D F}{8598}$ $\quad D = \frac{8598\ V}{\pi F}$	$A = \frac{D F^2}{2.53 \times 10^9}$ $\quad D = \frac{2.53 \times 10^9\ A}{F^2}$
$V = \frac{93,580\ A}{F}$ $\quad A = \frac{V F}{93,580}$	Where: A = RMS Acceleration (g) V = RMS Velocity (mm/sec) D = Peak-Peak Displacement (microns) F = Frequency (CPM)

FIGURE 1
CONTOURS OF EQUAL SEVERITY AND CONVERSION FORMULAS

For example, some users still acquire displacement spectra for most all of their machinery because "this is the way it has been done for many years". Figure 1 shows that while displacement does a good job on (and is the most sensitive parameter to) low frequency measurements predominantly below 600 CPM, it does not adequately detect problems which are higher frequency in nature such as rolling element bearing and gear wear. For example, assume a belt-driven blower with a nominal 3600 RPM motor, having a serious bearing defect frequency amplitude of .30 in/sec (peak) at 60,000 CPM, as well as a .30 in/sec amplitude at running speed (1X RPM). If peak-peak displacement were used to evaluate this machine, the equations in Figure 1 show that the .30 in/sec level at 60,000 CPM would correspond to a deceptively low amplitude of only .095 mil at

© Copyright 2001 Technical Associates Of Charlotte, P.C.

60,000 CPM; while the running speed vibration would be approximately 1.59 mils at 3600 CPM. Since normal alarms for this machine would be set at approximately 2 mils, the 3600 RPM running speed vibration would readily be visible in a linear amplitude spectrum, while that at the 60,000 CPM bearing frequency would not. This would give one a false sense of security concerning his bearing health. He may not even notice the presence of the bearing defect peak, and therefore, would not even attempt to determine its source. On the other hand, velocity spectra would clearly display the bearing frequency peak and would show it at just as high in amplitude as that at operating speed in this case (but in reality, of much greater problem severity than that at 1X RPM).

For another example, even though displacement would be the best indicator of unbalance or misalignment on a machine running only 300 RPM, if one's primary interest were rolling element bearing condition, velocity would again be the best parameter to employ. This is commonly the case with paper machines and other large, low-speed machinery.

On the other hand, velocity spectra have their limitations as well. For example, assume a common centrifugal air compressor running at 3580 RPM and having a 344 tooth bullgear. This machine would have a fundamental gear mesh frequency (GMF) of approximately 1,231,500 CPM (20,525 Hz). Experience with these machines proves, that not only must one evaluate amplitude at the fundamental gear mesh frequency, but at least at the second and third GMF harmonic as well. A good conditioned, well aligned set of gears will normally have a level of approximately 6 g (peak) at 2X GMF at approximately 2,463,000 CPM (41,050 Hz). If these gears were to develop definite problems increasing the amplitude 10 times higher to 60 g at 2X GMF, this would correspond to a peak velocity of only .089 in/sec (and an even more deceptive peak-peak displacement of only .00070 mil).

Therefore, the best indicator of problems which generate forcing frequencies in these high frequency regions, particularly above 300,000 CPM, is acceleration.

However, velocity spectra will prove to be the best indicator of a large majority of problems likely on about 80% to 90% of rotating machinery. Therefore, the emphasis of this article will be on specification of peak velocity spectral alarm bands (again, one can easily convert peak velocity overall amplitudes shown in Table II and spectral alarm band amplitudes shown in Table III to RMS simply by multiplying levels by a factor of .707). In fact, the great majority of instruments available actually make RMS measurements, and "convert" them to so-called peak velocity (or peak acceleration) by simply multiplying the amplitude measurement for each frequency by 1.414 ($\sqrt{2}$).

Similar bands can be specified for either displacement or acceleration for those machines requiring these parameters. Then, it will be important to take into account how displacement and acceleration vary with the frequency (see formulas in Figure 1). For example, assume that when using velocity, one were going to set the alarm level for 1X RPM at .30 in/sec, while setting the alarms at 2X and 3X RPM at one half this amount, or .15 in/sec. Since displacement varies directly with velocity, but inversely with frequency (see Figure 1), if he sets the 1X RPM displacement alarm at 6 mils, he will need to set the 2X RPM alarm level at only 1/4 that at 1X RPM (1.5 mil) and the 3X RPM alarm at only 1/6 that at 1X RPM (1 mil).

On the other hand, given the same machine, if one wished to convert from velocity to acceleration bands, he would need to make the relationship of one harmonic to another in a manner opposite that for displacement. For example, if the velocity alarms for 2X and 3X RPM have been set at one half that at 1X RPM, he will need to set the acceleration alarm at 2X RPM at the same level as that at 1X RPM, but set the 3X alarm at 50% higher than that at 1X RPM. This is due to the fact that acceleration varies directly with both velocity and frequency (see formula in Figure 1).

© Copyright 2001 Technical Associates Of Charlotte, P.C.

7.13 REVIEW OF PROBLEMS DETECTABLE BY VIBRATION ANALYSIS

An essential need in specifying effective spectral alarm bands is a firm understanding of what problems are detectable by vibration analysis, how they are detected, and, if detected, how severe they are. Much research has been performed and much continues today on how to evaluate such things as balance condition, alignment, bearing health, gear health, electrical condition, etc. **"Illustrated Vibration Diagnostic Chart"** in Table I represents the best understanding to date of the author on how these problems are best diagnosed, based on much field experience and research of a wide range of articles which have been written on the subject. The list of references researched for this Diagnostic Chart should give an idea of the study which was required to prepare these tables (Reference nos. 2, 3, 4, 5, 6, 9, 11, 12, 15, 16, 17, 20, 21, 22, 23, 25, 26, 27, 29, 30, 31, 34, 35, 36, and 38).

There are several key items included in Table I. First, the plots under "TYPICAL SPECTRUM" column reveal invaluable information about the source of the problem:

1. Which frequencies are present in spectrum and how do they relate to machine operating speed (1X RPM)?

2. What are the amplitudes of each peak?

3. How do the frequency peaks relate to one another? (i.e., "2X RPM is much higher than 1X RPM"; "there is a large peak at 7.43X RPM"; "there are large number of operating speed harmonics present"; "there are high amplitude sidebands around gear mesh frequency"; etc.)

As its column name implies, "TYPICAL SPECTRUM" is meant to be a representative signature for each problem listed in Table I. These spectra are not intended to be all inclusive. For example, referring to "REMARKS" for the Angular Misalignment problem, note that while the typical spectrum shows high amplitude 1X RPM and 2X RPM peaks in the axial direction, the discussion shows that it is not unusual for either 1X, 2X or 3X RPM to dominate the whole spectrum. Similarly, this can occur with either radial misalignment or a cocked bearing. In addition, it is not unusual for a machine to have two or more problems at any one time. For example, if a machine simultaneously had both mechanical looseness and rotor unbalance, they each would be indicated in its spectra which would likely show high 1X RPM, in addition to multiple running speed harmonics.

The next column in Table I is entitled "PHASE RELATIONSHIP". Information on phase is provided for several of the problem sources listed. Amplitude reveals how much something is vibrating. Frequency relates how many cycles occur per unit of time. Phase completes the picture by showing just how the machine is vibrating. Of great importance, phase is a powerful tool in helping differentiate which of several problem sources are dominant. For example, there are a large number of problems that generate vibration at 1X and 2X RPM. Using phase, one learns how the machine is vibrating, and in the process, helps zero in on just which problem is present. For example, Table I shows:

1. Force (or "static") unbalance is evidenced by nearly identical phase in the radial direction on each bearing of a machine (outboard and inboard horizontals are in phase; outboard and inboard verticals are in phase).

2. Couple unbalance shows approximately a 180° out-of-phase relationship when comparing the outboard and inboard horizontal, or outboard and inboard vertical direction phase on the same machine rotor.

3. Angular misalignment is indicated by approximately a 180° phase difference across the coupling, with measurements in the axial direction.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

4. Parallel misalignment causes radial direction phase across the coupling to be approximately 180° out of phase with respect to one another.

5. Bent shaft causes axial phase on the same shaft of a machine to approach 180° difference when comparing measurements on the outboard and inboard bearing of the same rotor.

6. Resonance is shown by exactly a 90° phase change at the point when the forcing frequency coincides with a natural frequency, and approaches a full 180° phase change when the machine passes through the natural frequency (depending on the amount of damping present).

7. Rotor rub causes significant, instantaneous changes in phase.

8. Mechanical looseness usually causes phase to be unsteady, with widely differing measurements from one time to the next. The phase measurement may noticeably differ every time you start up the machine, particularly if the rotor itself is loose and rotates on the shaft a few degrees with each startup.

Often, even though phase measurement capability is now offered by most data collectors, users do not use this powerful tool. If not used, this will severely limit the diagnostic capabilities of any program.

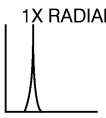
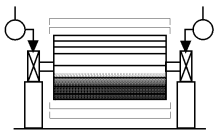
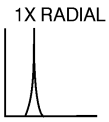
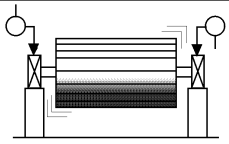
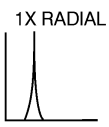
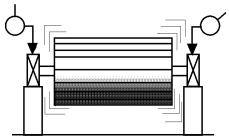
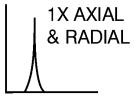
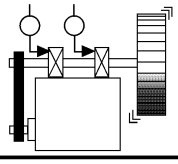
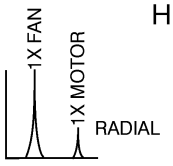
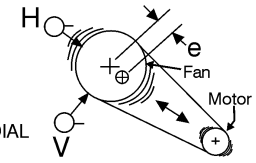
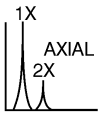
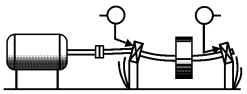
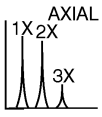
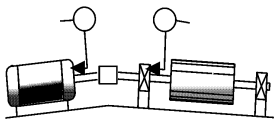
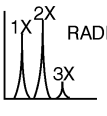
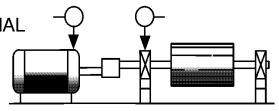
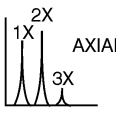
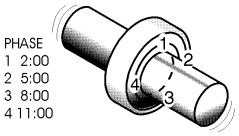
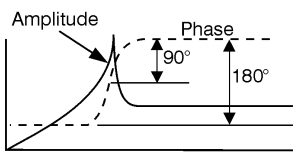
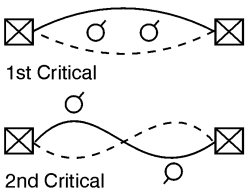
Note that “PHASE RELATIONSHIP” is illustrated in each of the first 8 problems of Table I since it is primarily with these problems that phase can be used to differentiate which problem(s) dominate. Phase is then discussed in many of the remarks for the remaining problems in Table I.

Finally, a “REMARKS” column is included in Table I to provide further explanatory information on machine problem symptoms and diagnostics. For example, there is a warning under the remarks column for the “bent shaft” problem source to be sure and account for transducer orientation when taking axial phase measurements.

It is hoped that this diagnostic chart will help users in diagnosing a wide variety of machine problems. Further information is now being researched and field tested which may soon be added to the diagnostic chart.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

TABLE I
ILLUSTRATED VIBRATION DIAGNOSTIC CHART

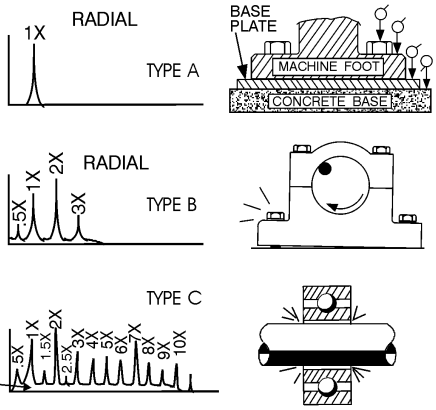
PROBLEM SOURCE	TYPICAL SPECTRUM	PHASE RELATIONSHIP	REMARKS
MASS UNBALANCE **A. FORCE UNBALANCE**			Force Unbalance will be in-phase and steady. Amplitude due to unbalance will increase by the square of speed below first rotor critical (a 3X speed increase = 9X higher vibration). 1X RPM always present and normally dominates spectrum. Can be corrected by placement of only one balance correction weight in one plane at Rotor center of gravity (CG). Approx. 0° phase difference should exist between OB & IB horizontals, as well as between OB & IB verticals. Also, approx. 90° phase difference between horizontal & vertical readings usually occurs on each bearing of unbalanced rotor (±30°).
B. COUPLE UNBALANCE			Couple Unbalance results in 180° out-of-phase motion on same shaft. 1X RPM always present and normally dominates spectrum. Amplitude varies with square of increasing speed below first rotor critical speed. May cause high axial vibration as well as radial. Correction requires placement of balance weights in at least 2 planes. Note that approx. 180° phase difference should exist between OB & IB horizontals, as well as between OB & IB verticals. Also, approx. a 90° difference between the horizontal & vertical phase readings on each bearing usually occurs (±30°)
C. DYNAMIC UNBALANCE			Dynamic Unbalance is the dominant type of unbalance found and is a combination of both force and couple unbalance. 1X RPM dominates the spectrum, and truly requires 2 plane correction. Here, the radial phase difference between outboard and inboard bearings can range anywhere from 0° to 180°. However, the horizontal phase difference should closely match the vertical phase difference, when comparing outboard and inboard bearing measurements (±30°). Secondarily, if unbalance predominates, roughly a 90° phase difference usually results between the horizontal and vertical readings on each bearing (±40°).
D. OVERHUNG ROTOR UNBALANCE			Overhung Rotor Unbalance causes high 1X RPM in both Axial and Radial directions. Axial readings tend to be in-phase whereas radial phase readings might be unsteady. However, the horizontal phase differences will usually match the vertical phase differences on the unbalanced rotor (±30°). Overhung rotors have both force and couple unbalance, each of which will likely require correction. Thus, correction weights will most always have to be placed in 2 planes to counteract both force and couple unbalance.
ECCENTRIC ROTOR			Eccentricity occurs when center of rotation is offset from geometric centerline of a pulley, gear, bearing, motor armature, etc. Largest vibration occurs at 1X RPM of eccentric component in a direction thru centerlines of the two rotors. Comparative horizontal and vertical phase readings usually differ either by 0° or by 180° (each of which indicate straight-line motion). Attempts to balance eccentric rotors often result in reducing vibration in one radial direction, but increasing it in the other radial direction (depending on amount of eccentricity).
BENT SHAFT			Bent shaft problems cause high axial vibration with axial phase differences tending towards 180° on the same machine component. Dominant vibration normally occurs at 1X if bent near shaft center, but at 2X if bent near the coupling. (Be careful to account for transducer orientation for each axial measurement if you reverse probe direction.) Use dial indicators to confirm bent shaft.
MISALIGNMENT **A. ANGULAR MISALIGNMENT**			Angular Misalignment is characterized by high axial vibration, 180° out-of-phase across the coupling. Typically will have high axial vibration with both 1X and 2X RPM. However, not unusual for either 1X, 2X or 3X to dominate. These symptoms may also indicate coupling problems as well. Severe angular misalignment may excite many 1X RPM harmonics. Unlike Mechanical Looseness Type 3, these multiple harmonics do not typically have a raised noise floor on the spectra.
B. PARALLEL MISALIGNMENT			Offset Misalignment has similar vibration symptoms to Angular, but shows high radial vibration which approaches 180° out-of-phase across coupling. 2X often larger than 1X, but its height relative to 1X is often dictated by coupling type and construction. When either Angular or Radial Misalignment becomes severe, they can generate either high amplitude peaks at much higher harmonics (4X-8X), or even a whole series of high frequency harmonics similar in appearance to mechanical looseness. Coupling type and material will often greatly influence the entire spectrum when misalignment is severe. Does not typically have raised noise floor.
C. MISALIGNED BEARING COCKED ON SHAFT			Cocked Bearing will generate considerable axial vibration. Will cause Twisting Motion with approximately 180° phase shift top to bottom and/or side to side as measured in axial direction on same bearing housing. Attempts to align coupling or balance the rotor will not alleviate problem. Bearing usually must be removed and correctly installed.
RESONANCE			Resonance occurs when a Forcing Frequency coincides with a System Natural Frequency, and can cause dramatic amplitude amplification, which might result in premature, or even catastrophic failure. This may be a natural frequency of the rotor, but can often originate from support frame, foundation, gearbox or even drive belts. If a rotor is at or near resonance, it can be almost impossible to balance due to the great phase shift it experiences (90° at resonance; nearly 180° when passes thru). Often requires changing natural frequency to a higher or lower frequency. Natural Frequencies do not generally change with a change in speed which helps facilitate their identification (unless on a large plain bearing machine or on a rotor which has significant overhang).

© COPYRIGHT 1996 - TECHNICAL ASSOCIATES OF CHARLOTTE, P.C. R-0894-4

TABLE I
ILLUSTRATED VIBRATION DIAGNOSTIC CHART

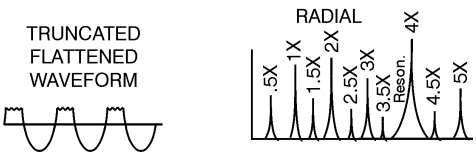
PROBLEM SOURCE	TYPICAL SPECTRUM	PHASE RELATIONSHIP	REMARKS

MECHANICAL LOOSENESS



Mechanical Looseness is indicated by either Type A, B or C vibration spectra.
Type A is caused by Structural looseness/weakness of machine feet, baseplate or foundation; also by deteriorated grouting, loose hold-down bolts at the base; and distortion of the frame or base (i.e., soft foot). Phase analysis may reveal approx. 90° to 180° phase difference between vertical measurements on bolt, machine foot, baseplate or base itself.
Type B is generally caused by loose pillowblock bolts, cracks in frame structure or in bearing pedestal.
Type C is normally generated by improper fit between component parts which will cause many harmonics due to nonlinear response of loose parts to dynamic forces from rotor. Causes a truncation of time waveform and a raised noise floor in the spectrum. Type C is often caused by a bearing liner loose in its cap, a bearing loose and turning on its shaft, excessive clearance in either a sleeve or rolling element bearing, a loose impeller on a shaft, etc. Type C Phase is often unstable and may vary widely from one measurement to next, particularly if rotor shifts position on shaft from one startup to next. Mechanical Looseness is often highly directional and may cause very different readings when comparing levels at 30° increments in radial direction all the way around one bearing housing. Also, note that looseness will often cause subharmonic multiples at exactly 1/2 or 1/3X RPM (.5X, 1.5X, 2.5X, etc.).

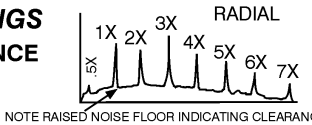
ROTOR RUB



Rotor Rub produces similar spectra to Mechanical Looseness when rotating parts contact stationary components. Rub may be either partial or throughout the entire shaft revolution. Usually generates a series of frequencies, often exciting one or more resonances. Often excites integer fraction subharmonics of running speed (1/2, 1/3, 1/4, 1/5,...1/n), depending on location of rotor natural frequencies. Rotor rub can excite many high frequencies (similar to wide-band noise when chalk is drug along a blackboard). It can be very serious and of short duration if caused by shaft contacting bearing babbitt. A full annular rub throughout an entire shaft revolution can induce "reverse precession" with the rotor whirling at critical speed in a direction opposite shaft rotation (inherently unstable which can lead to catastrophic failure).

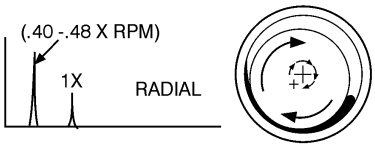
JOURNAL BEARINGS

A. WEAR/CLEARANCE PROBLEMS



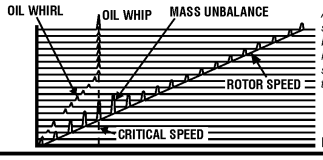
Latter stages of journal bearing wear are normally evidenced by presence of whole series of running speed harmonics (up to 10 or 20). Wiped journal bearings often will allow high vertical amplitudes compared to horizontal, but, may show only one pronounced peak at 1X RPM. Journal bearings with excessive clearance may allow a minor unbalance and/or misalignment to cause high vibration which would be much lower if bearing clearances were set to spec.

B. OIL WHIRL INSTABILITY



Oil Whirl instability occurs at .40 - .48X RPM and is often quite severe. Considered excessive when amplitude exceeds 40% of bearing clearances. Oil Whirl is an oil film excited vibration where deviations in normal operating conditions (attitude angle and eccentricity ratio) cause oil wedge to "push" shaft around within bearing. Destabilizing force in direction of rotation results in a whirl (or forwards precession). Oil Whirl is unstable since it increases centrifugal forces which increase whirl forces. Can cause oil to no longer support shaft and can become unstable when whirl frequency coincides with a rotor natural frequency. Changes in oil viscosity, lube pressure and external preloads can affect oil whirl.

C. OIL WHIP INSTABILITY

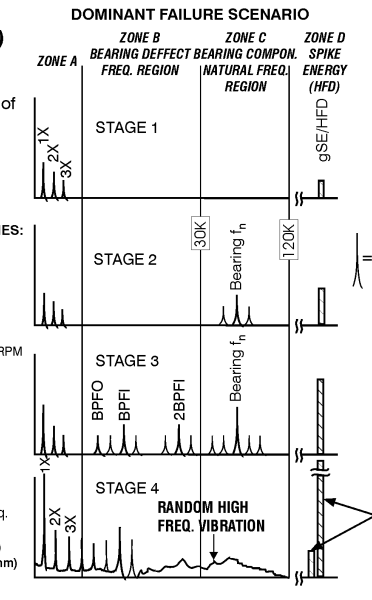


A spectral map showing Oil Whirl becoming Oil Whip instability as shaft speed passes thru 2X critical.

Oil Whip may occur if machine operated at or above 2X rotor critical frequency. When rotor brought up to twice critical speed, whirl will be very close to rotor critical and may cause excessive vibration that oil film may no longer be capable of supporting. Whirl speed will actually "lock onto" rotor critical and this peak will not pass through it even if machine is brought to higher and higher speeds. Produces a lateral forward precessional subharmonic vibration at rotor critical frequency. Inherently unstable which can lead to catastrophic failure.

ROLLING ELEMENT BEARINGS

(4 Failure Stages)



f_n = Natural Frequencies of Installed Bearing Components and Support Structure

BEARING DEFECT FREQUENCIES:

$$BPFI = \frac{N_b}{2}\left(1+\frac{B_d}{P_d}\cos\theta\right) \times RPM$$

$$BPFO = \frac{N_b}{2}\left(1-\frac{B_d}{P_d}\cos\theta\right) \times RPM$$

$$BSF = \frac{P_d}{2B_d}\left[1-\left(\frac{B_d}{P_d}\right)^2(\cos\theta)^2\right] \times RPM$$

$$FTF = \frac{1}{2}\left(1-\frac{B_d}{P_d}\cos\theta\right) \times RPM$$

Where:
BPFI = Inner Race Frequency
BPFO = Outer Race Frequency
BSF = Ball Spin Frequency
FTF = Fund. Train (Cage) Freq.
N_b = Number of Balls or Rollers
B_d = Ball/Roller Diameter (in or mm)
P_d = Bearing Pitch Diameter (in or mm)
θ = Contact Angle (degrees)

4 ROLLING ELEMENT BEARING FAILURE STAGES

STAGE 1: Earliest indications of bearing problems appear in ultrasonic frequencies ranging from about 250,000 - 350,000 Hz; later, as wear increases, usually drops to approximately 20,000 - 60,000 Hz (1,200,000 - 3,600,000 CPM). These are frequencies evaluated by Spike Energy (gSE), HFD(g) and Shock Pulse (dB). For example, spike energy may first appear at about .25 gSE in Stage 1 (actual value depending on measurement location and machine speed). Acquiring high frequency enveloped spectra confirms whether or not bearing is in Failure Stage 1.

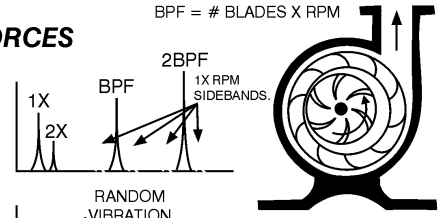
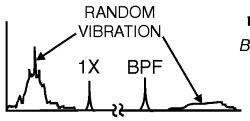
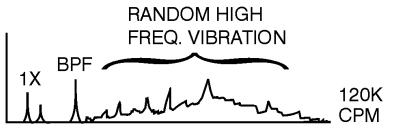
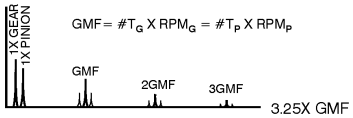
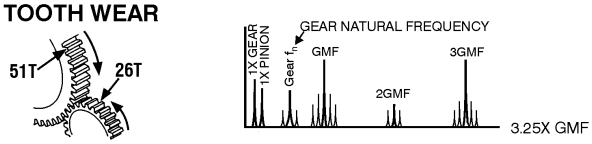
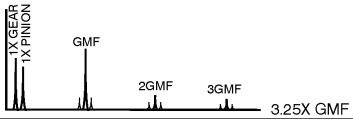
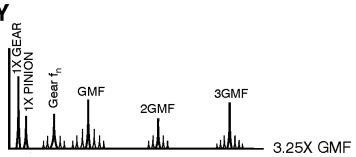
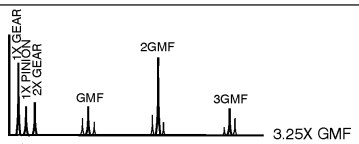
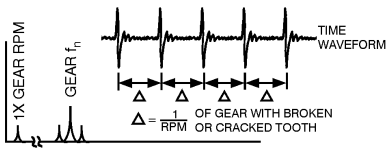
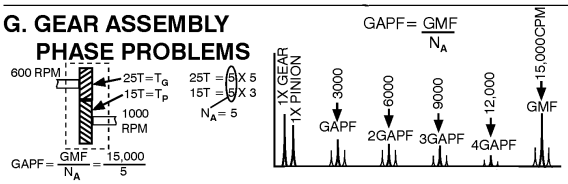
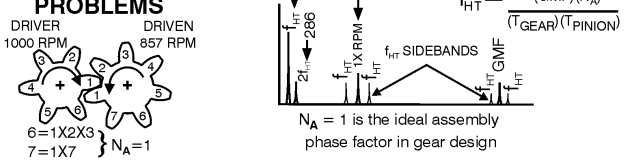
STAGE 2: Slight bearing defects begin to "ring" bearing component natural frequencies (f_n) which predominantly occur in 30K - 120K CPM range. Such natural frequencies may also be resonances of bearing support structures. Sideband frequencies appear above and below natural frequency peak at end of Stage 2. Overall spike energy grows (for example, from .25 to .50 gSE).

STAGE 3: Bearing defect frequencies and harmonics appear. When wear progresses, more defect frequency harmonics appear and number of sidebands grow, both around these and bearing component natural frequencies. Overall spike energy continues to increase (for example, from .5 to over 1 gSE). Wear is now usually visible and may extend throughout periphery of bearing, particularly when many well formed sidebands accompany bearing defect frequency harmonics. High frequency demodulated and enveloped spectra help confirm Stage III. ***Replace bearings now! (independent of bearing defect frequency amplitudes in vibration spectra).***

STAGE 4: Towards the end, amplitude of 1X RPM is even effected. It grows, and normally causes growth of many running speed harmonics. Discrete bearing defect and component natural frequencies actually begin to "disappear" and are replaced by random, broadband high frequency "noise floor". In addition, amplitudes of both high frequency noise floor and spike energy may in fact decrease; but just prior to failure, spike energy and HFD will usually grow to excessive amplitudes.

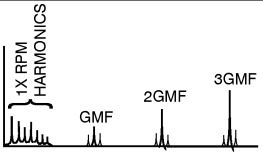
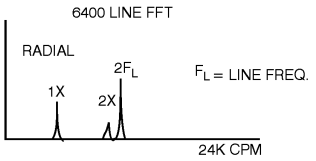
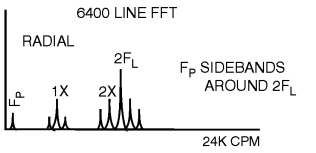
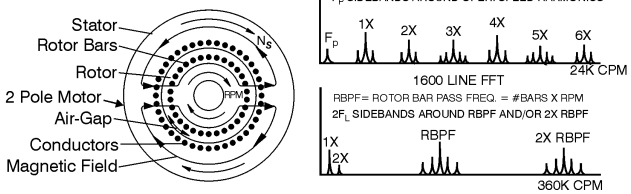
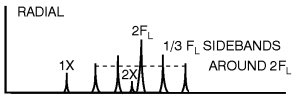
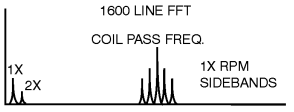
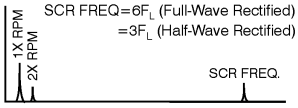
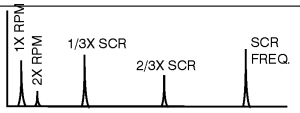
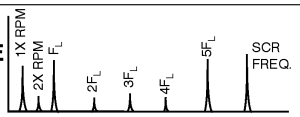
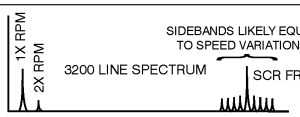
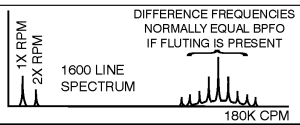
© COPYRIGHT 1996 - TECHNICAL ASSOCIATES OF CHARLOTTE, P.C. R-0894-4

TABLE I
ILLUSTRATED VIBRATION DIAGNOSTIC CHART

PROBLEM SOURCE	TYPICAL SPECTRUM	REMARKS
HYDRAULIC AND AERODYNAMIC FORCES		
A. BLADE PASS & VANE PASS		Blade Pass Frequency (BPF) = No. of Blades (or Vanes) X RPM. This frequency is inherent in pumps, fans and compressors, and normally does not present a problem. However, large amplitude BPF (and harmonics) can be generated in pump if gap between rotating vanes and stationary diffusers is not equal all the way around. Also, BPF (or harmonic) sometimes can coincide with a system natural frequency causing high vibration. High BPF can be generated if impeller wear ring seizes on shaft, or if welds fastening diffuser vanes fail. Also, high BPF can be caused by abrupt bends in pipe (or duct), obstructions which disturb flow, damper settings or if pump or fan rotor is positioned eccentrically within housing.
B. FLOW TURBULENCE		Flow Turbulence often occurs in blowers due to variations in pressure or velocity of the air passing thru the fan or connected ductwork. This flow disruption causes turbulence which will generate random, low frequency vibration, typically in the range of 50 to 2000 CPM. If surging occurs within a compressor, random broadband high frequency vibration can occur. Excessive turbulence can also excite broadband high frequency.
C. CAVITATION		Cavitation normally generates random, higher frequency broadband energy which is sometimes superimposed with blade pass frequency harmonics. Normally indicates insufficient suction pressure (starvation). Cavitation can be quite destructive to pump internals if left uncorrected. It can particularly erode impeller vanes. When present, it often sounds as if "gravel" is passing thru pump. Cavitation is usually caused by insufficient inlet flow. Can occur during one survey, and be absent the next survey (if changes in suction valve settings are made).
GEARS		
A. NORMAL SPECTRUM		Normal Spectrum shows Gear & Pinion Speeds, along with Gear Mesh Frequency (GMF) and very small GMF harmonics. GMF harmonics commonly will have running speed sidebands around them. All peaks are of low amplitude, and no natural frequencies of gears are excited. F_{MAX} recomended at 3.25X GMF (minimum) when # teeth are known. If tooth count is not known, set F_{MAX} at 200X RPM on each shaft.
B. TOOTH WEAR		Key indicator of Tooth Wear is excitation of Gear Natural Frequency (f_n), along with sidebands around it spaced at the running speed of the bad gear. Gear Mesh Frequency (GMF) may or may not change in amplitude, although high amplitude sidebands and number of sidebands surrounding GMF usually occur when wear is noticeable. Sidebands may be better wear indicator than GMF frequencies themselves. Also, high amplitudes commonly occur at either 2XGMF or at 3XGMF (esp. 3XGMF), even when GMF amplitude is acceptable.
C. TOOTH LOAD		Gear Mesh Frequencies are often very sensitive to load. High GMF amplitudes do not necessarily indicate a problem, particularly if sideband frequencies remain low level, and no gear natural frequencies are excited. Each Analysis should be performed with system at maximum operating load for meaningful spectral comparisons.
D. GEAR ECCENTRICITY AND BACKLASH		Fairly high amplitude sidebands around GMF harmonics often suggest gear eccentricity, backlash, or non-parallel shafts which allow the rotation of one gear to "modulate" either the GMF amplitude or the running speed of the other gear. The gear with the problem is indicated by the spacing of the sideband frequencies.Also, 1X RPM level of eccentric gear will normally be high if eccentricity is the dominant problem. Improper backlash normally excites GMF harmonics and Gear Natural Frequency, both of which will be sidebanded at 1X RPM. GMF amplitudes will often decrease with increasing load if backlash is the problem.
E. GEAR MISALIGNMENT		Gear Misalignment almost always excites second order or higher GMF harmonics which are sidebanded at running speed. Often will show only small amplitude 1X GMF, but much higher levels at 2X or 3X GMF. Important to set F_{MAX} high enough to capture at least 3 GMF harmonics. Also, sidebands around 2XGMF will often be spaced at 2X RPM. Note that sideband amplitudes often are not equal on left and right side of GMF and GMF harmonics due to the tooth misalignment. Causes uneven wear pattern.
F. CRACKED/BROKEN TOOTH		A Cracked or Broken Tooth will generate a high amplitude at 1X RPM of this gear only in the time waveform, plus it will excite gear natural frequency (f_n) sidebanded at its running speed. It is best detected in Time Waveform which will show a pronounced spike every time the problem tooth tries to mesh with teeth on the mating gear. Time between impacts (Δ) will correspond to 1/RPM of gear with the problem. Amplitudes of Impact Spikes in Time Waveform often will be 10X to 20X higher than that at 1X RPM in the FFT!
G. GEAR ASSEMBLY PHASE PROBLEMS	 GAPF= 3000 CPM = 0.20X GMF **(FRACTIONAL GMF)**	Gear Assembly Phase Freq. (GAPF) can result in Fractional Gear Mesh Frequencies (if N_A>1). It literally means (T_G/N_A) gear teeth will contact (T_P/N_A) pinion teeth and will generate N_A wear patterns, where N_A in a given tooth combination equals the product of prime factors common to the number of teeth on the gear and pinion (N_A = Assembly Phase Factor). GAPF (or harmonics) can show up right from the beginning if there were manufacturing problems. Also, its sudden appearance in a periodic survey spectrum can indicate damage if contaminate particles pass through the mesh, resulting in damage to the teeth in mesh at the time of ingestion just as they enter and leave meshing or that gears have been reoriented.
H. HUNTING TOOTH PROBLEMS	 f_{HT}= $\frac{(6X\ 1000)(1)}{(6)(7)}$ = $\frac{1000}{7}$ = 143 CPM (One Pulse Per 7 Pinion Revolutions)	Hunting Tooth Frequency (f_{HT}) occurs when faults are present on both the gear and pinion which might have occurred during the manufacturing process, due to mishandling, or in the field. It can cause quite high vibration, but since it occurs at low frequencies predominately less than 600 CPM, it is often missed. A gear set with this tooth repeat problem normally emits a "growling" sound from the drive. The maximum effect occurs when the faulty pinion and gear teeth both enter mesh at the same time (on some drives, this may occur only 1 of every 10 to 20 revolutions, depending on the f_{HT} formula). Note that T_{GEAR} and T_{PINION} refer to number of teeth on the gear and pinion, respectively. N_A is the Assembly Phase Factor defined above. Will often modulate both GMF and Gear RPM peaks.

© COPYRIGHT 1996 - TECHNICAL ASSOCIATES OF CHARLOTTE, P.C. R-0894-4

TABLE I
ILLUSTRATED VIBRATION DIAGNOSTIC CHART

PROBLEM SOURCE	TYPICAL SPECTRUM	REMARKS
GEARS (CONTINUED)		
I. LOOSE BEARING FIT		Excessive Clearance of bearings supporting the gears can not only excite many running speed harmonics, but will often cause high amplitude response at GMF, 2GMF and/or 3GMF. These high GMF amplitudes are actually a response to, and not the cause of, looseness within the bearings supporting the gearing. Such excessive clearance can be caused either by extensive bearing wear or by improper bearing fit onto the journal during installation. Left uncorrected, it can cause excessive gear wear and damage to other components.
AC INDUCTION MOTORS		
A. STATOR ECCENTRICITY, SHORTED LAMINATIONS OR LOOSE IRON		Stator problems generate high vibration at 2X line frequency ($2F_L$). Stator eccentricity produces uneven stationary air gap between rotor and stator which produces very directional vibration. Differential Air Gap should not exceed 5% for induction motors and 10% for synchronous motors. Soft foot and warped bases can produce an eccentric stator. Loose iron is due to stator support weakness or looseness. Shorted stator laminations can cause uneven, localized heating which can distort the stator itself. This produces thermally-induced vibration which can significantly grow with operating time causing stator distortion and static air gap problems.
B. ECCENTRIC ROTOR (Variable Air Gap) F_L = Electrical Line Freq. N_S = Synch. Speed = $\frac{120F_L}{P}$ F_S = Slip Freq. = N_S – RPM F_P = Pole Pass Freq. = F_S X P P = #Poles		Eccentric Rotors produce a rotating variable air gap between the rotor and stator which induces pulsating vibration (normally between $2F_L$ and closest running speed harmonic). Often requires "zoom" spectrum to separate $2F_L$ and running speed harmonic. Eccentric rotors generate $2F_L$ surrounded by Pole Pass frequency sidebands (F_P), as well as F_P sidebands around running speed. F_P appears itself at low frequency (Pole Pass Frequency = Slip Frequency X #Poles). Common values of F_P range from about 20 to 120 CPM (0.3 - 2.0 Hz). Soft foot or misalignment often induces a variable air gap due to distortion (actually a mechanical problem; not electrical).
C. ROTOR PROBLEMS		Broken or Cracked rotor bars or shorting rings; bad joints between rotor bars and shorting rings; or shorted rotor laminations will produce high 1X running speed vibration with pole pass frequency sidebands (F_P). In addition, these problems will often generate F_P sidebands around the second, third, fourth and fifth running speed harmonics. Loose or open rotor bars are indicated by 2X line freq. ($2F_L$) sidebands surrounding Rotor Bar Pass Frequency (RBPF) and/or its harmonics (RBPF = Number of Bars X RPM). Often will cause high levels at 2X RBPF, with only a small amplitude at 1X RBPF. Electrically induced arcing between loose rotor bars and end rings will often show high levels at 2X RBPF (with $2F_L$ sidebands); but little or no increase in amplitudes at 1X RBPF.
D. PHASING PROBLEM (Loose Connector)		Phasing problems due to loose or broken connectors can cause excessive vibration at 2X Line Freq. ($2F_L$) which will have sidebands around it spaced at 1/3 Line Freq. ($1/3\ F_L$). Levels at $2F_L$ can exceed 1.0 in/sec if left uncorrected. This is particularly a problem if the defective connector is only sporadically making contact. Loose or broken connectors must be repaired to prevent catastrophic failure.
AC SYNCHRONOUS MOTORS **(Loose Stator Coils)**		Loose stator coils in synchronous motors will generate fairly high vibration at Coil Pass Freq. (CPF) which equals the number of stator coils X RPM (#Stator Coils = #Poles X #Coils/Pole). The Coil Pass Frequency will be surrounded by 1X RPM sidebands. Synchronous motor problems may also be indicated by high amplitude peaks at approx. 60,000 to 90,000 CPM, accompanied by $2F_L$ sidebands. Take at least one spectrum up to 90,000 CPM on each motor bearing housing.
DC MOTORS AND CONTROLS **A. NORMAL SPECTRUM**		Many DC Motor and Control Problems can be detected by vibration analysis. Full-wave rectified, motors (6 SCR's) generate a signal at 6X Line Frequency ($6F_L$=360 Hz=21,600 CPM); while half-wave rectified DC motors (3 SCR's) generate 3X Line Freq. ($3F_L$=180 Hz=10,800 CPM). The SCR firing Frequency is normally present in a DC Motor Spectrum, but at low amplitude. Note the absence of other peaks at multiples of F_L.
B. BROKEN ARMATURE WINDINGS, GROUNDING PROBLEMS OR FAULTY SYSTEM TUNING		When DC Motor spectra are dominated by high levels at SCR or 2X SCR, this normally indicates either Broken Motor Windings or Faulty Tuning of the Electrical Control System. Proper tuning alone can lower vibration at SCR and 2X SCR significantly if control problems predominate. High amplitudes at these frequencies would normally be above approximately .10 in/sec, peak at 1 X SCR and about .04 in/sec at 2 X SCR Firing Freq.
C. FAULTY FIRING CARD OR BLOWN FUSE		When one firing card fails to fire, then 1/3 of power is lost, and can cause repeated momentary speed changes in the motor. This can lead to high amplitudes at 1/3X and 2/3X SCR Frequency (1/3X SCR Freq. = 1X F_L for half-wave rectified, but 2X F_L for a full-wave rectified SCR). Caution: Card/SCR configuration should be known before troubleshooting motor (#SCR's, #Firing Cards, etc.).
D. FAULTY SCR, SHORTED CONTROL CARD, LOOSE CONNECTIONS AND/OR BLOWN FUSE		Faulty SCR's, Shorted Control Cards and/or Loose Connections can generate noticeable amplitude peaks at many combinations of line frequency (F_L) and SCR firing frequency. Normally, 1 bad SCR can cause high levels at F_L and/or $5F_L$ in 6 SCR motors. The point to be made is that neither F_L, $2F_L$, $4F_L$ nor $5F_L$ should be present in DC Motor spectra.
E. FAULTY COMPARITOR CARD		Faulty Comparitor Cards cause problems with RPM fluctuation or "hunting". This causes a constant collapsing and regenerating of the magnetic field. These sidebands often approximate the RPM fluctuation and require a high resolution FFT to even detect them. Such sidebands could also be due to generation and regeneration of the magnetic field.
F. ELECTRICAL CURRENT PASSAGE THRU DC MOTOR BEARINGS		Electrically-induced Fluting is normally detected by a series of difference frequencies with the spacing most often at the outer race defect frequency (BPFO), even if such fluting is present on both the outer and inner races. They most often show up in a range centered at about 100,000 to 150,000 CPM. A 180K CPM spectrum with 1600 lines is recommended for detection with measurements on both the OB and IB DC motor bearings.

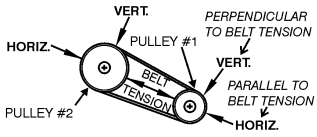
© COPYRIGHT 1996 - TECHNICAL ASSOCIATES OF CHARLOTTE, P.C. R-0894-4

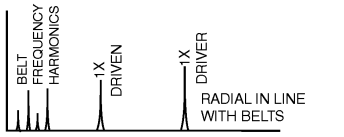
TABLE I
ILLUSTRATED VIBRATION DIAGNOSTIC CHART

PROBLEM SOURCE	TYPICAL SPECTRUM	REMARKS

BELT DRIVE PROBLEMS

A. WORN, LOOSE OR MISMATCHED BELTS





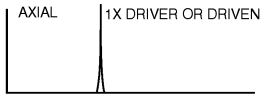
PITCH DIAM$_1$ X RPM$_1$ = PITCH DIAM$_2$ X RPM$_2$

BELT FREQ. = $\frac{3.142 \text{ X PULLEY RPM X PITCH DIAM.}}{\text{BELT LENGTH}}$

TIMING BELT FREQ. = BELT FREQ. X #BELT TEETH
= PULLEY RPM X #PULLEY TEETH

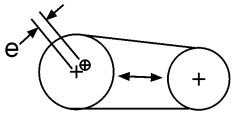
Belt frequencies are below the RPM of either the motor or the driven machine. When they are worn, loose or mismatched, they normally cause 3 to 4 multiples of belt frequency. Often 2X belt freq. is the dominant peak. Amplitudes are normally unsteady, sometimes pulsing with either driver or driven RPM. On timing belt drives, wear or pulley misalignment is indicated by high amplitudes at the Timing Belt Frequency. Chain drives will indicate problems at Chain Pass Frequency which equals #Sprocket Teeth X RPM.

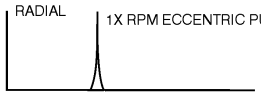
B. BELT/PULLEY MISALIGNMENT



Misalignment of pulley produces high vibration at 1X RPM predominantly in the axial direction. The ratio of amplitudes of driver to driven RPM depends on where the data is taken, as well as on relative mass and frame stiffness. Often with pulley misalignment, the highest axial vibration on the motor will be at fan RPM, or vice versa. Can be confirmed by phase measurements by setting Phase Filter at RPM of pulley with highest axial amplitude; then compare phase at this particular frequency on each rotor in the axial direction.

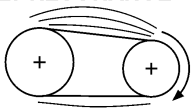
C. ECCENTRIC PULLEYS

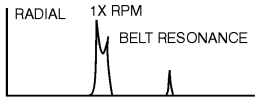




Eccentric pulleys cause high vibration at 1X RPM of the eccentric pulley. The amplitude is normally highest in line with the belts, and should show up on both driver and driven bearings. It is sometimes possible to balance eccentric pulleys by attaching washers to taper-lock bolts. However, even if balanced, the eccentricity will still induce vibration and reversible fatigue stresses in the belt. Pulley eccentricity can be confirmed by phase analysis showing horizontal & vertical phase differences of nearly 0° or 180°.

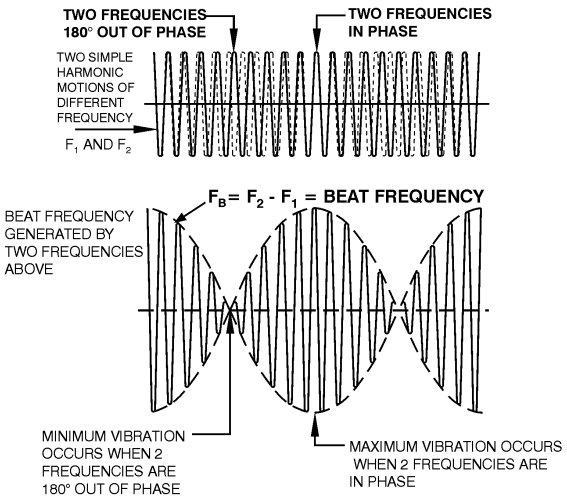
D. BELT RESONANCE





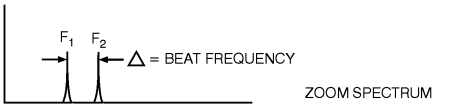
Belt Resonance can cause high amplitudes if the belt natural frequency should happen to approach, or coincide with, either the motor or driven RPM. Belt natural frequency can be altered by changing either the belt tension, belt length or cross section. Can be detected by tensioning and then releasing belt while measuring the response on pulleys or bearings. However, when operating, belt natural frequencies will tend to be slightly higher on the tight side and lower on the slack side.

BEAT VIBRATION

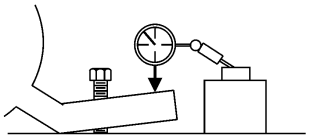


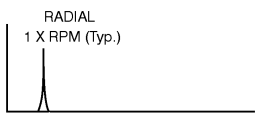
A Beat Frequency is the result of two closely spaced frequencies going into and out of synchronization with one another. The wideband spectrum normally will show one peak pulsating up and down. When you zoom into this peak (lower spectrum below), it actually shows two closely spaced peaks. The difference in these two peaks (F_2-F_1) is the beat frequency which appears itself in the wideband spectrum. The beat frequency is not commonly seen in normal frequency range measurements since it is inherently low frequency, usually ranging from only approximately 5 to 100 CPM.

Maximum vibration will result when the time waveform of one frequency (F_1) comes into phase with the waveform of the other frequency (F_2). Minimum vibration occurs when waveforms of these two frequencies line up 180° out of phase.



SOFT FOOT, SPRUNG FOOT AND FOOT-RELATED RESONANCE





"***Soft Foot***" occurs when a machine's foot or frame deflects greatly when a hold-down bolt is loosened to hand tightness, causing the foot to rise more than approximately .002 - .003 inch. This does not always cause a great vibration increase. However, it can do so if the soft foot affects alignment or motor air gap concentricity.

"***Sprung Foot***" can cause great frame distortion, resulting in increased vibration, force and stress in the frame, bearing housing, etc. This can occur when a hold-down bolt is forceably torqued down on the sprung foot in an attempt to level the foot.

"***Foot-Related Resonance***" can cause dramatic amplitude increases from 5X to 15X or more, as compared with that when the bolt (or combination of bolts) is loosened to hand tightness. When tight, this bolt can notably change the natural frequency of the foot or machine frame itself.

Soft Foot, Sprung Foot or Foot-Related Resonance most often affects vibration at 1X RPM, but can also do so at 2X RPM, 3X RPM, 2X line frequency, blade pass frequency, etc. (particularly Foot-Related Resonance).

© COPYRIGHT 1996 - TECHNICAL ASSOCIATES OF CHARLOTTE, P.C. R-0894-4

During the years, our company has had the opportunity of analyzing a diverse array of both process and utility machinery ranging from very small, precision, high-speed spindles to large, slow moving machines. In addition, this work has been performed for a wide array of industry types. This has given us invaluable exposure which has been greatly beneficial when opportunities have arisen for setting up condition monitoring programs for these same clients.

Through the years, we have developed in-house vibration criteria specifically for the purpose of setting up these condition monitoring databases. Some of the criteria we have developed is included in Table II. Note that Table II includes overall peak velocity criteria for measurements taken on the machine structure. Importantly, levels specified in Table II are not meant to be final, concrete numbers, but are intended to be a starting point when nothing is known about a machine other than its nameplate data, machine type and mounting. Later, after taking actual measurements on each point of each machine, these levels are individually reviewed and adjusted as needed. This refinement procedure is discussed later in the paper and examples are given illustrating the procedure.

Note that each of four "ratings" are provided in Table II including "GOOD", "FAIR", "ALARM 1" and "ALARM 2". After review of all spectra captured on a machine, if no problems are found, the first two columns ("GOOD" and "FAIR") are offered to give the client a general feel for the overall condition of each machine based on the highest overall level measured on his machine. However, even if the highest overall on a machine might remain within the "GOOD" range, it is still possible for the machine to be in alarm, depending on what frequencies were generated, the amplitudes of those frequencies, and the problem source(s) generating these frequencies. That is where the spectral alarm bands come into play to ferret out the "apparently good condition" machines from those that truly have a problem. Corrective actions should be taken on those machines having vibration exceeding "ALARM 1"; while those exceeding "ALARM 2" are felt to be exposed to such high levels as to render potentially catastrophic failure (therefore, demanding immediate attention). Amplitudes listed in Table II were developed by calculating both the average level and standard deviation of large quantities of diverse types of machinery over a period of approximately 20 years in carrying out condition monitoring programs. Then, "ALARM 1" overall levels were calculated by summing the average level plus 3 times the standard deviation [see Equations (2) through (4B)]. Final statistical overall levels were then rounded to the nearest ".025" level (that is, a level of .318 would be rounded off to .325 in/sec). Finally "ALARM 2" levels were determined by increasing "ALARM 1" levels by 50%.

Importantly, not only do the overall levels specified in Table II serve as an overall alarm given in the PMP database, but also they are used as direct input for specifying alarm levels for each specific spectral band in the section which will follow. Of course, if this overall is later refined after making several measurements on a machine or on a group of machines, the spectral band alarm levels themselves will also have to be adjusted as well.

7.15 SPECIFICATION OF SPECTRAL ALARM LEVELS AND FREQUENCY BANDS USING TABLE III

Table III provides the tabulated procedure on how to originally specify spectral alarm bands for various machine types and configurations using those types of condition monitoring software systems which allow the spectrum to be broken up into 6 individual bands. Each of these bands can be set at any span of frequencies, and at any alarm level for each individual band as chosen by the user. Therein lies both the strength of spectral alarm bands, and paradoxically, their major weakness if the user himself does not know where each frequency span should be specified, nor how high to set each one of the band alarm levels. Therefore, the express purpose of this section is to provide the condition monitoring software user with the capability, not only of originally setting up a PMP database using proper spectral alarm bands, but also to help him refine his database on which he might have been taking measurements for several years. Several years ago, our company made a detailed in-house study on how to specify these bands. At the conclusion

© Copyright 2001 Technical Associates Of Charlotte, P.C.

of this study, we elected to develop a written, tabular procedure on how to properly specify them. Since that time, we have helped a number of clients set up bands on their specific machinery in their particular industry. In so doing, we have continued to learn more and more about how to best use them, and have "polished" and refined our techniques several times. In addition, much study went into preparation of Table III as can be seen by the list of references (see Reference nos. 3, 6, 8, 10, 11, 12, 13, 14, 18, 23, 26, 28, 29, 32, and 37).

Importantly, please note that the procedures specified in Table III assume casing measurements of peak velocity (in/sec) using instruments which measure RMS and "convert" them to peak levels by electronic multiplication of amplitudes by 1.414. This now includes most of the data collectors in use in the United States. Also, Table III specifies spectral bands whose alarm levels are compared to the total power within the band (so-called "Power Bands"). Please refer to the section entitled "Two Types of Spectral Alarm Bands"). Although Table III applies to peak velocity amplitudes, the reader can modify it for RMS simply by multiplying amplitudes by .707. Then, if he wishes to have them expressed in metric units (mm/sec), he can multiply these RMS in/sec amplitudes by a factor of 25.4 and rounding them to the nearest appropriate metric level.

Table III shows how spectral alarm bands are set up for a number of machine types and configurations. Cases A and B are for both the driver and driven components of general rotating machines which are outfitted with rolling element and sleeve bearings, respectively. Cases C and D specify high frequency measurement points which are to be taken on gearbox housings in close proximity to each gear mesh, and which are essential to evaluate the health and alignment of gearing. Case C assumes one knows the number of gear teeth, while Case D shows how to specify alarm bands for gearboxes where the number of teeth is unknown. Cases E and F are special points with the purpose of detecting potential motor electrical problems. The point specified by Case E is intended to detect the first and second harmonic rotor bar pass frequencies (number of bars X RPM), whereas the point specified by Case F attempts to separate mechanical from electrical vibration sources, particularly in the vicinity of machine operating speed, electrical synchronous frequency (60 Hz), and twice synchronous frequency (120 Hz). Case G covers how to specify alarm bands for centrifugal compressors, blowers and pumps. Cases H and I have been added to this paper in its fourth edition. Case H covers DC motors and controls while Case I encompasses machine tool spindles.

Importantly, the specification procedure outlined in Table III applies to general process and utility machinery such as centrifugal pumps, blowers, motors, forced-draft fans, induction-draft fans, motor/generator sets, centrifugal air compressors, refrigeration chillers, vacuum pumps, boiler feed pumps, gearboxes, etc. These specs do not apply to more specialized machine types such as reciprocating or rotary screw compressors; diesel engines; gas turbines; large turbine/generators; exciters; lobe-type rotary blowers; pulverizers; etc. Normally, spectral bands for these machine types have to be "custom-designed" for each set or grouping of them, and even then, often require the capture of several sets of data before one can begin to establish meaningful alarm bands.

For example, lobe-type rotary blowers (i.e., "Roots Blowers") present a real problem to the user who attempts to specify one all-encompassing set of alarm bands. They are offered in a wide range of sizes and configurations. Often, even after several surveys are conducted on these machines, the user may have difficulty in adequately specifying alarm bands since even identically sized and driven rotary blowers still can exhibit unique sets of vibration spectra (23). In reality, only 6 spectral alarm bands cannot adequately address these machines. They need approximately 10 to 12 bands (or more) to adequately cover them.

However, if the user is given the assignment of specifying spectral alarm bands for his plant, either when originally setting up its database or after several years of data have been captured (without adequate alarm bands), the procedure given in Table III should cover a large percentage of his machines. Before entering Table III, the user should identify his particular machine type and refer

© Copyright 2001 Technical Associates Of Charlotte, P.C.

to Table II to find the alarm level of overall vibration for this machine. This will be used as direct input into the spectral alarm band specs of Table III. If his particular machine type is not included in Table II, the user should either refer to the manufacturer of his machine, other similar vibration severity charts, or use alarm levels for another machine type listed in Table II which most closely resembles his machine.

Please refer to the entries under the first column of Table III. The "BAND LOWER FREQUENCY" specifies at what frequency each band should begin, whereas "BAND HIGHER FREQUENCY" shows where each band should end (for example, "from 60 to 1000 CPM"). In general, no gaps should be left between bands, nor should bands overlap one another (although some analysts using power bands sometimes extend one band from the beginning to the end of a complete spectrum in order to have the system calculate the "Spectral Overall Level", and then compare this to the overall level provided separately by their instrument). Next, the column entry entitled "BAND ALARM" specifies how high to set the alarm level of each band.

Notice that many of the cases described in Table III have the "BAND LOWER FREQUENCY" set at 1% F_{MAX} rather than at 0 CPM. The reason for this is that data collectors and spectrum analyzers most always have built-in "noise" within the first 1 to 3 FFT lines, particularly when data from an accelerometer are electronically integrated to velocity. In fact, some instruments have been known to display "peaks" with so-called "amplitudes" over 2.0 in/sec within these first 3 FFT lines. If F_{MAX} is properly specified, the first 2 to 3 FFT lines will almost always be contaminated with such electronic and/or integration "noise". Therefore, Band 1 will never begin within these first 3 lines in Table III.

Each of the cases specify the maximum frequency (F_{MAX}) which is always given along with the case title. Therefore, each case will tell where to set both the frequency range and alarm level of each band, and will describe what each band covers (i.e., bearing defect frequencies, gear mesh frequencies, etc.). Case A will be discussed in detail to illustrate the alarm band specification technique, whereas only highlights of each remaining case will be given. Then, several examples will follow the discussions to further illustrate how these techniques should be applied.

Case A - General Rolling Element Bearing Machine Without Rotating Vanes:
(Motors, Gearbox Lower Frequency Measurements, etc.)

Case A applies to a wide range of general rotating process and utility machines which are outfitted with rolling element bearings (ball, roller or needle bearings). Before entering Table III, refer to Table II to obtain the alarm level of overall peak velocity for your machine type. Then, determine the type of rolling element bearing. For common rolling element bearings, Case A specifies a spectrum with a maximum frequency (F_{MAX}) of approximately 50X RPM (for example, for a nominal speed of 1800 RPM, set F_{MAX} at approximately 90,000 CPM). However, for tapered roller bearings (Timken cup and cone arrangement, or equivalent) or for spherical roller bearings, Case A specifies a maximum frequency of approximately 60X RPM. The reason for the higher F_{MAX} for these bearing types is the fact that, with their particular geometries, they inherently have higher calculated rolling element bearing defect frequencies. Also note that if the speed is below 1700 RPM, F_{MAX} must be set higher than 50X RPM (as seen in notes of Case A). The reason for this is to ensure that the spectra designed for this machine will detect a rolling element bearing in only the second of four failure stages through which it will normally pass rather than waiting late in the life of the bearing before problems are detected. Referring to Table I for "Rolling Element Bearings", note that the natural frequencies of bearing components will be excited during this second stage. Since these natural frequencies normally range from 30,000 to 90,000 CPM for most bearings, it is important to keep F_{MAX} sufficiently high to detect these when excited (bearing natural frequencies may range as high as 120,000 to 150,000 for specialty rolling element bearings such as aircraft bearings, small bore bearings, etc.).

© Copyright 2001 Technical Associates Of Charlotte, P.C.

Please note that it is not necessary to specify F_{MAX} at exactly 50X or 60X RPM, but it should be somewhere in this vicinity (certainly not less than 45X RPM). If one sets F_{MAX} too low, it can cause a spectrum to completely miss potentially serious developing bearing wear, particularly during earlier stages. On the other hand, if F_{MAX} is set too high, this can result in poor frequency resolution which can cause the user to misdiagnose problems, since he does not have sufficiently precise frequency resolution to properly identify such frequency components as true running speed harmonics versus bearing defect frequencies, or vibration transmitting from adjacent machines. Also, if one sets F_{MAX} too high, this can cause potentially valuable information on subsynchronous vibration to be "buried" on the left-hand side of the spectrum. In general, the rule of thumb is to keep F_{MAX} as low as you can "without missing anything important".

Referring to Case A in Table III, note that each one of the bands has a specific purpose and zone of coverage. For example, Band 1 ranges from subsynchronous vibration (below 1X RPM) up through operating speed. Bands 2 and 3 cover 2X and 3X RPM, respectively. Band 4 will include fundamental bearing defect frequencies for most rolling element bearings. Similarly, Bands 5 and 6 will include bearing defect frequency harmonics, as well as natural frequencies of bearing components for most common rolling element bearings.

Now, referring back to Table III, note that Band 1 extends from 1% of F_{MAX} to a frequency at 1.2X RPM. In the case of the example 1800 RPM machine shown in Figure 10 having F_{MAX} at 90,000 CPM, Band 1 would extend from 900 to 2160 CPM. The Band 1 alarm spec calls for 90% of the overall level. Thus, if the overall alarm were .300 in/sec (from Table II), then the Band 1 alarm would be set at .270 in/sec for this machine. Similarly, Table III specifies the frequency range of Band 2 to extend from 1.2 to 2.2X RPM (in the 1800 RPM case, this would extend from approximately 2160 to 3960 CPM). The Band 2 alarm spec calls for 30% of the overall alarm (thus for the example .300 in/sec overall, Band 2 would be set at .090 in/sec). Finally, Bands 3 through 6 are specified similarly.

Case B - General Sleeve Bearing Machine Without Rotating Vanes:
(Sleeve Bearing Motors, Gearbox Lower Frequency Measurements, etc.)

Case B is similar to Case A, but is for general machines outfitted with sleeve bearings. Incidentally, if a sleeve bearing motor is driving a rolling element machine, Case A (rolling element) would be used for the driven machine points, whereas Case B (sleeve bearing) would be applied to the points on the motor. However, refer to Cases G thru I if the driven component is a centrifugal machine, DC motor or machine tool spindle.

Notice that F_{MAX} for these sleeve bearing machines is set only at 20X RPM as compared to 50X up to 120X RPM on rolling element bearing machines which inherently have much higher frequency spectra. In addition, some potentially serious problems can occur at subsynchronous frequencies on sleeve bearings including such things as oil whirl and oil whip. Therefore, this subsynchronous band needs to have good frequency resolution and must be closely watched.

Band 1 covers only the subsynchronous vibration in Case B while Bands 2, 3 and 4 include 1X, 2X and 3X RPM peaks, respectively (see Table III). Band 5 covers the range from 4X through 10X RPM while Band 6 extends from 10.5X RPM to F_{MAX}. Here again, the highest alarm level specified for any of the bands in Case B will be that at 1X RPM (Band 2). On the other hand, little amplitude is allowed in Band 6 even though it covers about 50% of the entire spectrum since only insignificant vibration should occur in this region if problems are not present, particularly if this machine is not a gearbox or connected to a gearbox.

Case C - Gearbox High Frequency Points with Known Number of Teeth:

Gearboxes require two sets of measurements on the same points due to the fact that many gear problems are detected at very high frequencies as compared to vibration due to such problems as

© Copyright 2001 Technical Associates Of Charlotte, P.C.

unbalance, misalignment, etc. Therefore, one set of measurements should be taken on the gearbox using either Case A or B, depending on whether the gearbox is outfitted with rolling element (Case A) or sleeve (Case B) bearings. Then, a second set of measurements should be taken at various gearbox points close to each mesh, with F_{MAX} on this second measurement then set at 3.25X gear mesh frequency as shown in Table III. Very commonly, gearboxes may show low amplitudes at the fundamental gear mesh frequency (GMF), but may display very high amplitudes at 2X and/or 3X GMF. In addition, looseness is sometimes evidenced at one-half harmonics of gear mesh frequency, up to 2.5X GMF. Therefore, the maximum frequency is set at 3.25X GMF in order to allow for capture of gear mesh and accompanying sideband frequencies up thru 3X GMF.

Please note in Table III that spectra with 1600 to 3200 lines of resolution are recommended for these high frequency measurements. The reason for this is to allow 1X RPM sidebands to be displayed with good resolution around gear mesh frequency harmonics, not only for the high speed pinion, but also for the lower speed gear. Such high resolution spectra will also be recommended for Case D (when the tooth count is unknown).

A complete example illustrating specification of spectral alarm bands for a 2-stage speed increaser gearbox driving a compressor is given in Figure 11. Note the setups for both the lower frequency measurements (i.e., positions 3HI Axial and 3HI Horizontal) in Figure 11.

Please note the caution under Case C to keep in mind that a requirement to set F_{MAX} at 3.25X GMF may cause one to specify a maximum frequency that is not necessarily greater than the transducer frequency specifications, but can easily approach the natural frequency of the transducer mounting itself, causing errors in amplitude measurements. That is, when a transducer is mounted on a machine, it just creates another “spring/mass” in the system. The natural frequency of this spring/mass depends on how the transducer is mounted on the machine (stud, magnet, hand-held or extension probe). Stud mounting provides the highest natural frequency and, therefore, allows the highest measurable frequency with little or no deviation in amplitude readout. If forcing frequencies (such as gear mesh frequencies) are present close to the mounting natural frequency, considerable amplification can occur causing error in the amplitude readout, but not in the frequency. On the other hand, if forcing frequencies are above the mounting natural frequency, they can result not only in deviation in amplitude readout, but can also cause phase error since this transducer/mount system will experience almost a 180° difference in phase when it passes through resonance.

However, if this is kept in mind by the user, he can still take data at fairly high frequencies, being aware that amplitude levels may not be absolute. In any case, if they are repeatable, they can at least be trended; and, since frequency information remains correct, will likely allow the user to detect potential problems. If they are not repeatable, he will have to try a different transducer or method of mounting the original transducer.

Referring again to Case C in Table III, each of the frequency ranges are specified in terms of GMF multiples (for example, Band 2 extends from .75X GMF to 1.25X GMF). Here again, band alarms are set in terms of overall alarm percentage.

Importantly, if the gearbox has more than one set of individual meshes, as in the case of a double or a triple reduction unit, each set of high frequency points will need to employ the gear mesh frequency that applies at that particular measurement point. For example, if point A were close to the input gear mesh having a 100,000 CPM GMF and point B was at the output near a second mesh with a 25,000 CPM GMF, the high frequency point A would use the 100,000 CPM when setting up its bands (setting F_{MAX} at 325,000 CPM), whereas point B would employ the 25,000 CPM GMF (setting its F_{MAX} at approximately 81,250 CPM). This will be further illustrated in a gearbox example to be given later (Figure 8).

© Copyright 2001 Technical Associates Of Charlotte, P.C.

Case D - Gearbox High Frequency Points with Unknown Number of Teeth:

Unfortunately, in most programs, the number of teeth in the great majority of gearboxes is unknown. In many cases, even the operating speeds of intermediate gears are unknown. However, in spite of this, one can set up effective spectral alarm bands which can be used until the true number of teeth are confirmed (however, when the tooth count is found, the spectral alarm bands should be respecified as per Case C). Referring to Case D in Table III, note that a maximum frequency of 200X shaft RPM will apply to each high frequency gearbox point. Note that the shaft speed at each particular measurement point will be used in specifying frequency ranges for each of the 6 bands. For example, if the input speed at point A was 1000 RPM and the output speed at point B was about 200 RPM, F_{MAX} at point A would be set at 1000 RPM X 200 (200,000 CPM) while that at point B would be set at 200 RPM X 200 (40,000 CPM).

In many plants, both the number of teeth and intermediate speeds are unknown in many multi-stage gearboxes. One approach to this problem of determining what the gear mesh frequencies are might be to acquire several sets of spectra on the gearbox and compare them to the "TYPICAL SPECTRUM" shown in Table I, Case E for "Gear Misalignment" (note that it shows both GMF and 2GMF, each of which are sidebanded at 1X RPM). If two or three harmonics of a high frequency fundamental are found (for example, fundamental at 40 to 60X RPM), it is possible that these are gear mesh frequencies, particularly if they each have 1X RPM sidebands. However, one must keep in mind that this same signature pattern could be caused by another problem (for example, rolling element bearing frequency harmonics at, say, 5X and 10X inner race frequency).

Therefore, Table III, Case D suggests another approach if the number of teeth and intermediate speeds are unknown. In these cases, one normally knows at least the gearbox ratio, and therefore, the input and output speeds. The note in Case D shows how to handle this case in which equal speed increment steps are assumed until one knows more about the intermediate shaft speeds. For example, if all you knew were the input speed, output speed and gear ratio, use the following formula as a start until you know more:

$$\text{Speed Increment Factor} = (\text{Gear Ratio})^{1/m}$$

Where m = number of separate gear meshes

For example, for a triple reduction gearbox with:

Input RPM = 3594, Assumed Speed Increment = ?
Output RPM = 230, Assumed Interm. #1 RPM = ?
Gear Ratio = 15.625, Assumed Interm. #2 RPM = ?

Gear Ratio = 15.625; and 1/m = 1/3 = .3333
(3 meshes)

Thus, Speed Increment Factor = $(15.625)^{.3333}$ = 2.50
Assumed Interm. #1 Speed = 3594/2.50 = <u>1438 RPM</u>
Assumed Interm. #2 Speed = 1438/2.50 = <u>575 RPM</u>

Again, when intermediate shaft speeds are confirmed, use these speeds in Case D. And, when the tooth count is confirmed, change spectral band setups <u>immediately</u> back to those specified in Case C of Table III.

Like Case C, the same caution is given on keeping in mind the high frequency limits of the transducer and its mounting. Often, this requires one to stud mount or temporarily epoxy the transducers for these high frequency measurements.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

Case E - AC Induction Motor Electrical Rotor Bar Pass Frequency Point (single point usually taken on outboard motor bearing):

The specific purpose of this measurement on each motor is to detect the presence of 1X and 2X rotor bar pass frequencies which often are accompanied by 2X line frequency (7200 CPM) sidebands, 1X RPM sidebands, and even pole pass frequency sidebands (see Table I). The rotor bar pass frequency (RBPF) is equal to the number of rotor bars times motor RPM. High amplitudes at rotor bar pass frequencies suggest rotor bar looseness and/or rotor eccentricity, particularly when these frequencies are accompanied by the 2X line and/or pole pass frequency sidebands.

Please note that this data is only taken at one point on each motor (normally on the outboard horizontal housing). Notice that the maximum frequency for this point (F_{MAX}) is fixed at 360,000 CPM. Also, note that Band 1 begins at 30,000 CPM and incrementally takes 55,000 CPM steps in each succeeding band up to 360,000 CPM in Band 6 (independent of operating speed for this point which applies to 900 to 3600 RPM motors). Here again, recall that the standard points also taken on this motor (specified using either Case A or B, depending on the bearing type) will evaluate unbalance, misalignment, etc. The number of rotor bars in most all motors is rarely known, but normally ranges between about 35 to 95. Therefore, the F_{MAX} of 360,000 CPM should almost always encompass both the first and second harmonic rotor bar pass frequencies, even on 2-pole, nominal 3600 CPM motors. 1600 line spectra are recommended here with 8 to 16 spectrum averages. Since F_{MAX} is so high, even 16 averages of 1600 line spectra should require only about 7 to 10 seconds total. However, due to the high frequency, this might require permanent placement of a disk using a thin layer of high frequency epoxy adhesive in order to provide a dependable measurement mounting. Use of a high strength, rare-earth magnet is recommended in order to provide a good transducer mounting for this important electrical measurement.

Case F - AC Induction Motor Electrical Measurement Point (single point usually taken on inboard motor bearing):

The whole purpose of this single point on each motor is to (1) attempt to separate mechanical and electrical vibration frequencies, particularly in the area of 1X RPM, line frequency (3600 CPM or 60 Hz), and 2X line frequency (7200 CPM or 120 Hz); and (2) to detect the possible presence of pole pass frequency sidebands around running speed harmonics. Very often in predictive maintenance programs, the spectrum will show high vibration at a so-called frequency of 7200 CPM which might suggest electrical problems. However, unless one has the required frequency resolution to separate running speed harmonics from the electrical synchronous frequencies, he cannot truly detect the presence of either a mechanical or an electrical problem, its severity, and certainly not its cause (variable air gap, stator problems, etc.). This is due to the fact that with an F_{MAX} of 50X RPM, he cannot separate, for example, the 3580 RPM operating speed peak from the 3600 CPM line frequency.

Therefore, if one uses 3200 FFT lines of resolution and a 12,000 CPM F_{MAX}, he will likely be able to separate most of these mechanical and electrical frequencies, depending on the motor RPM. Note that 400 FFT lines with a 12,000 CPM F_{MAX} will result in a 30 CPM frequency resolution which means peaks must be at least 90 CPM apart to show two separate frequencies (2X frequency resolution X 1.5 Hanning Noise Factor). For example, if the speed of a 2-pole motor is approximately 3550 RPM, one would be able to separate running speed from 3600 CPM line frequency and 2X RPM (7100 CPM) from 2X line frequency (7200 CPM) using only 400 lines and a 12,000 CPM F_{MAX}. However, if the motor speed were higher in the range of 3590 RPM (10 CPM slip frequency X 2 poles = 20 CPM pole pass frequency, it will require 3200 lines of resolution to display both the running speed harmonics and pole pass sidebands (3200 lines with a span of 12,000 CPM will give a frequency resolution of 3.75 CPM and a bandwidth of 5.625 CPM allowing the analyst to see each set of frequencies). Due to the high resolution of 3200 lines and rather

© Copyright 2001 Technical Associates Of Charlotte, P.C.

small frequency span of 12,000 CPM, this will require 16 seconds for the first average. Therefore, it is recommended that 50% overlap processing be used taking 2 averages for this measurement (which will result in 16 seconds for the first average and 8 seconds for the second average, or a total of 24 seconds). However, the end result of this measurement is critical. It alone will allow the analyst to separate mechanical and electrical problems, plus detect potentially serious cracked or broken rotor bars (which he cannot even detect using 400 lines and an F_{MAX} of 50X RPM or so). It will also allow detection of subsynchronous belt defect frequencies on belt-driven machinery.

Case G - Centrifugal Compressors, Blowers and Pumps

Driven components require a different setup of spectral band alarms than those specified for the Driving components which are covered in Cases A and B (i.e., motors, turbines, gearboxes, etc.). This section will cover various centrifugal machine types including pumps, blowers and compressors. For example, the primary purpose for building special bands for centrifugal machines outfitted with rolling element bearings is to attempt to separate the blade pass frequency band from the bearing defect frequency band. The problem here is that amplitudes which would be acceptable at blade pass frequencies (BPF) would normally be excessive for a bearing defect frequency. If both the blade pass frequency and bearing frequencies coexist within the same band, it would be impossible to separate the alarm levels for these two unique sources. This will be discussed in following sections below.

Note that Types 1 and 2 cover measurements on centrifugal machines outfitted with rolling element bearings while Types 3 and 4 cover such machines with sleeve (or journal) bearings. Types 1 and 3 assume the number of impeller blades (or vanes) is known whereas Types 2 and 4 assume the analyst does not yet know the number of blades (when the number of blades is confirmed, replace the setups for Types 2 or 4 immediately with Type 1 or 3 (depending on whether the measurement is on a rolling element or on a sleeve bearing). Spectral alarm band setups are much more meaningful and effective when the number of impeller blades is confirmed.

Type 1 - Driven Centrifugal Component with Known Number of Vanes (or Blades) and Rolling Element Bearings:

Type 1 will cover driven centrifugal machines outfitted with rolling element bearings where the number of vanes (or blades) in a pump, fan or compressor is known. In these cases, it will be possible to set up a separate band to capture blade pass frequency (BPF), allowing a higher alarm for it than that for the bands containing bearing defect frequencies (BPFI, BPFO, etc.). This procedure is illustrated in Table III under Type 1 of Case G. Band 4 will include the fundamental blade pass frequency (BPF) as well as 1X RPM sidebands above and below BPF. This band will have an alarm level of 60% of the overall alarm. On the other hand, bearing frequency Bands 3 and 5 on either side of Band 4 will have much lower alarms as seen in Type 1 of Case G. Notice that Band 5 will likely capture not only lower harmonic bearing frequencies, but also harmonics of blade pass frequencies which might relate to flow pulsation problems.

Type 2A - Centrifugal Pumps with Unknown Number of Vanes and Rolling Element Bearings:

Type 2A covers pumps outfitted with rolling element bearings when the number of impeller vanes is not known. In this case, the frequency limits for the probable BPF in Band 4 are set to capture what should be blade pass frequency for roughly 60% to 80% of centrifugal pumps which often have 4 to 6 vanes. Of course, if this is not the case, Band 4 can be adjusted. In any case, when the number of vanes is found, replace the spectral alarm bands shown here with those given in Type 1. Notice in Type 2A that the Band 4 alarm is set at 60% of the overall alarm as in the case of Type 1. Here, the intention is to ensure that if fundamental bearing frequencies do occur within this band, amplitudes will not be allowed to become highly excessive before a potential bearing problem is detected. Fortunately, even if fundamental bearing frequencies do coexist with blade pass within Band 4, worn bearings typically will generate several harmonics of bearing defect frequencies exceeding the higher frequency of Band 4 where alarm levels will be much lower (see

© Copyright 2001 Technical Associates Of Charlotte, P.C.

Bands 5 and 6 in Type 2A of Case G). Figure 2 shows where measurement locations should be established on the bearing caps of a centrifugal pump. Take care not to mistake a seal or packing gland for a bearing.

Type 2B - Centrifugal Blowers & Compressors with Unknown Number of Blades and Rolling Element Bearings:

Spectral band setups in Type 2B show that experience has proven the Blade Pass Frequency amplitudes for most blowers and centrifugal compressors are typically lower than BPF amplitudes on a centrifugal pump (note the Band 4 alarm of 40% of the Overall alarm not to exceed .100 in/sec for these centrifugal machines versus levels of 60% of the overall not to exceed .185 in/sec for pumps). In addition, there are typically more blades on blower and compressor impellers than on pump impellers.

Thus, the "assumed" BPF for Type 2B is expected to be in the vicinity of 8X to 12X RPM. Of course, once the actual number of blades is ascertained, the analyst is instructed to immediately replace this Case 2B setup with that shown in Type 1 since a much better set of alarm bands will be employed in those cases where the BPF is confirmed. Figure 3 shows optimum locations for measurements on the bearing caps of centrifugal air compressors.

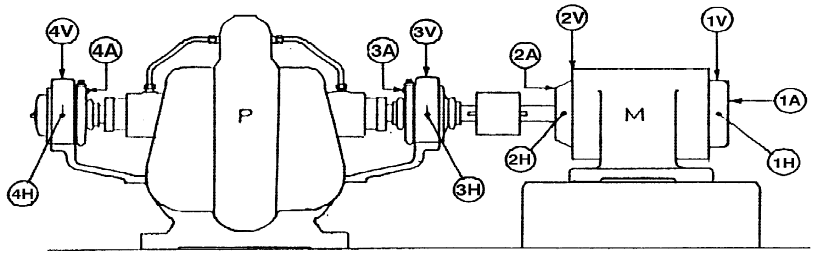


FIGURE 2 MEASUREMENT LOCATIONS ON CENTRIFUGAL PUMPS

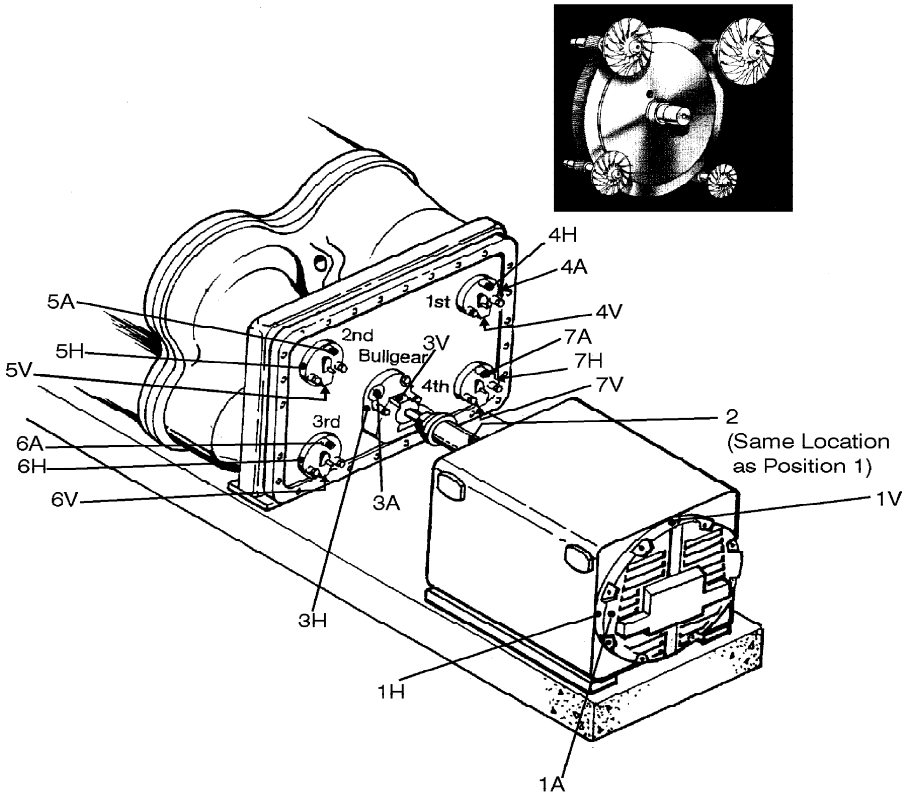


FIGURE 3 REQUIRED MEASUREMENT LOCATIONS ON CENTRIFUGAL AIR COMPRESSORS

© Copyright 2001 Technical Associates Of Charlotte, P.C.

Type 3 - Driven Centrifugal Component with Known Number of Vanes (or Blades) and Sleeve Bearings:

Types 3 and 4 differ from Types 1 and 2 in that these centrifugal machines are outfitted with sleeve (or journal) bearings rather than rolling element bearings. In this case, the maximum frequency will be set at only 20X RPM or 1.2X Blade Pass (whichever is greater), versus 50 to 120X RPM in the case of a unit outfitted with rolling element bearings. Here, Band 5 will capture the blade pass frequency (BPF) as well as 1X RPM sidebands above and below BPF. Band 5 will have an alarm set at 70% of the overall, versus only 30% of the overall alarm in the case of Bands 4 and 6 to its left and right. Here again, note that driven components require different spectral band alarm

Case H - DC Motors: Full-Wave and Half-Wave SCR Controlled Rectifier

DC motors controlled by silicon controlled rectifiers (SCR's) require special measurements in order to detect problems not only within the motors, but also those within the controls serving them (see "DC Motors and Controls" section of the **"Diagnostic Chart"** in Table I). The spectral setup parameters required to detect these faults are shown in Types 1, 2 and 3 of Section H in Table III. Each of these measurements require fine frequency resolution spectra in order to detect the faults within these machines and controls (3200 - 6400 lines). Figure 4 shows the general construction of a DC motor and where measurements are taken.

It should be noted that the spectral band setups shown in Case H assume a line frequency (F_L) of 60 Hz (3600 CPM). If the line frequency differs from 60 Hz, the frequency span of each band should likewise be adjusted. For example, if served by a line frequency of 50 Hz (3000 CPM), the analyst should center 600 CPM wide bands around 3000 CPM and its harmonics (i.e., 2700-3300 CPM in Band 1 for Types 1 and 2).

Note that Types 1, 2 and 3 are special measurements specifically used to detect electrical problems within DC motors and controls. These measurements should typically be taken only in the horizontal direction as per Case H of Table III. Standard spectral setups like those shown in Cases A and B should be taken in horizontal, vertical, and axial directions on both the outboard and inboard bearings of each DC motor in order to detect mechanical problems such as unbalance, misalignment, looseness, bearing problems, etc.

Type 1 - Full-Wave Rectified Motors (Measure on Commutator-End Bearing in the Horizontal Direction):

The purpose of this special measurement taken on the commutator end of full-wave rectified motors is to detect problems with armature windings, commutators, SCR's, firing cards, comparitor cards, fuses, etc. In order to detect these problems, one must use 6400 lines and 2 averages with 50% overlap processing. This will allow the analyst not only to detect 1X RPM sidebands around the SCR firing frequency, but also Δ RPM sidebands which would likely be present if a DC motor has comparitor card problems causing its speed to fluctuate. Note that the SCR firing frequency for a full-wave rectified motor is 21,600 CPM (360 Hz) assuming a line frequency (F_L) of 60 Hz. Looking at the spectral band setup shown in Type 1, note that amplitudes of only .02 in/sec are specified for Bands 1 thru 5. This is due to the fact that the first 5 line frequency harmonics should not be present in a DC motor served by SCR controls. On the other hand, the SCR firing frequency (6 F_L) is expected to be seen just as 2X line frequency (2 F_L) in an AC motor or blade pass frequency (BPF) in a pump. However, problems are indicated if the amplitude at SCR firing frequency exceeds the alarm level. Note that the alarm level of .08 in/sec shown in Band 6 (covering SCR firing frequency and 1X RPM sidebands) has been determined by sampling large quantities of DC motor spectra, statistically processing the amplitudes at this frequency and determining appropriate alarm levels which were noted when certain faults were found present with either DC motors or the controls serving them.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

Type 2 - Half-Wave Rectified Motors (Measure on Commutator-End Bearing in the Horizontal Direction):

This measurement is very similar to that shown for Type 1 with the exception that this spectral setup covers half-wave rectified motors where the SCR firing frequency is 10,800 CPM (180 Hz) assuming a line frequency (F_L) of 60 Hz. Looking at the spectral band setups for Type 2 in Table III, note that alarm levels of only .02 in/sec are specified for 1X line and 2X line frequencies since these frequencies should not be present within a SCR controlled DC motor; similarly, levels of only .015 in/sec are specified at 4X line and 5X line frequency for these half-wave rectified motors. Note that the alarm specified for the SCR firing frequency (3X line frequency for half-wave rectified motors) is again .08 in/sec as determined by comprehensive statistical compilations on half-wave rectified motors.

Type 3 - Special Measurement Point Required For Detection of Electrically-Induced Fluting on Full-Wave or Half-Wave Rectified DC Motors:

The primary purpose of this special measurement taken on both the outboard and inboard bearings of a DC motor is to detect possible electrically-induced fluting which is caused by electrical current flow through bearings. The **"Diagnostic Chart"**, Table I, shows that fluting is normally detected by a series of difference frequencies modulating a carrier frequency in the neighborhood of 100,000 to 150,000 CPM. These difference frequencies will most often occur at the outer race defect frequency (BPFO), but can occasionally appear at the inner race frequency (BPFI) as well. Note that if only standard measurements are taken on DC motors up to approximately 50X RPM, the presence of electrically-induced fluting will very likely not be detected. As the spectral criteria in Type 3 shows, a frequency range extending to 180,000 CPM (3000 Hz) should be specified, along with a frequency resolution of 3200 lines and at least 8 averages. Typically, this data should be taken in the horizontal direction on both the outboard and inboard bearings since such fluting can occur on either or both bearings. However, if a horizontal measurement would be distant from the bearing, whereas an axial measurement is either on or much closer to the bearing, the measurement should be taken in the axial direction due to the rather high frequency range required. In addition, the analyst should ensure the transducer is well mounted due to the rather high maximum frequency of 180,000 CPM. Alarm levels for each of the 6 bands are shown in Type 3 of Case H in Table III.

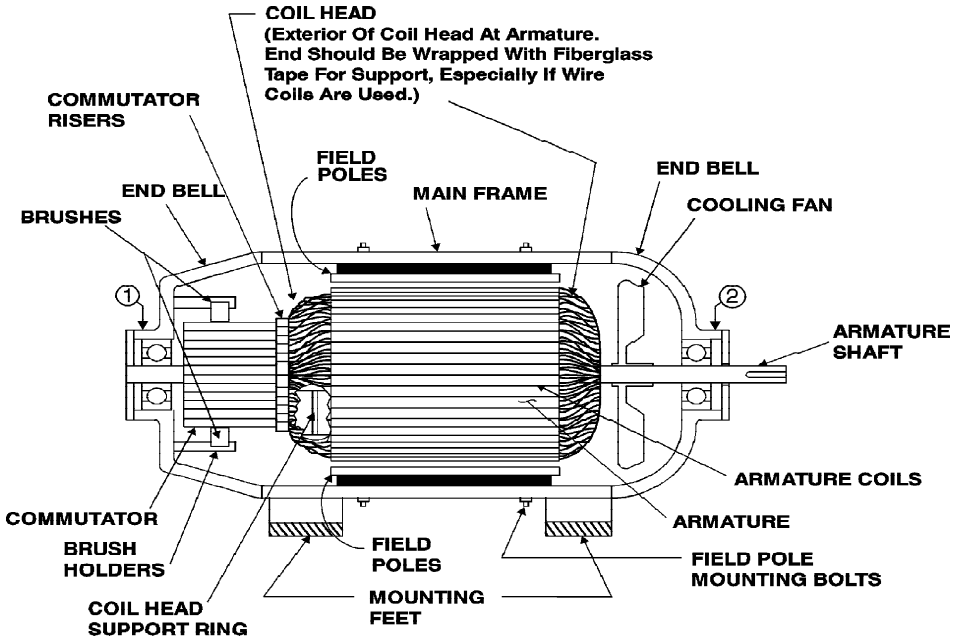


FIGURE 4 GENERAL DC MOTOR CONSTRUCTION (Ref. 41)

© Copyright 2001 Technical Associates Of Charlotte, P.C.

Case I - Machine Tool Spindles and Multi-Gear Heads (Ref. 40)

Machine tools are used to produce a wide variety of products. Machine tools are constructed of high quality components to maintain rigorous tolerances and to withstand significant stresses imposed upon them during machining processes. These machines typically have some of the lower vibration amplitudes in comparison with most all other machine types.

Table II includes overall vibration levels anticipated for machine tools. Note particularly the "Alarm 1" and "Alarm 2" levels specified for machine tool spindles. Three different grades are specified in Table II with the following "Alarm 1" amplitudes:

a.	Roughing Operations	.100 in/sec
b.	Machine Finishing Operations	.060 in/sec
c.	Critical Finishing Operrations	.040 in/sec

Figures 5 thru 9 include a variety of machine tool types (these drawings are extracted from Ref. 40). These include each of the following machine tool types:

SPINDLE TYPE	FIG. NO.
Precision Box Spindle	Fig. 5
Precision Cartridge-Type Spindle	Fig. 6
Precision Multi-Gear Head Spindle	Fig. 7
Precision Multi-Gear Head Gearbox	Fig. 7
Precision Grinding Wheel Head	Fig. 8
Precision Integral-Motor Spindle	Fig. 9

Following below are brief descriptions of each of the machine tools listed above, along with drawings of each spindle type covered:

Precision Box Spindles (Fig. 5) - Driven by belts or couplings and include foot-mounted or block design types. Precision Box Spindles may be used in, but not limited to, drilling, reaming, chamfering, spot fishing, countersinking, boring, milling, and single-wheel grinding operations. Figure 5 shows an example of a Precision Box Spindle.

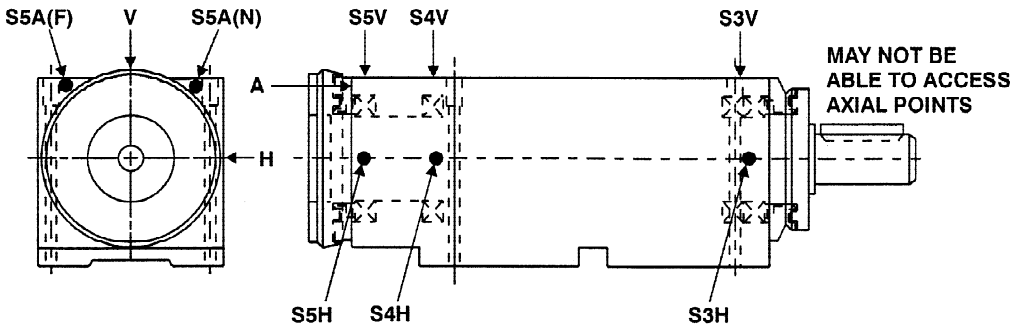


FIGURE 5
EXAMPLE OF CONSECUTIVE MEASUREMENT POINT LAYOUT ON PRECISION BOX SPINDLES
(REF. Setco 4300B Series Spindles, Catalog #181D, p. 9)

© Copyright 2001 Technical Associates Of Charlotte, P.C.

Precision Cartridge-Type Spindles (Fig. 6) - Driven by belts or couplings and include plain or flange design types. Precision Cartridge-Type Spindles may be used in, but not limited to, drilling, boring, milling, turning, and internal deep-hole grinding operations. Figure 6 shows an example Precision Cartridge-Type Spindle. Note that in cases where the outboard bearings are the only accessible bearings, horizontal, vertical and axial measurements should be taken on this outboard bearing location only (per spindle). See Figure 6.

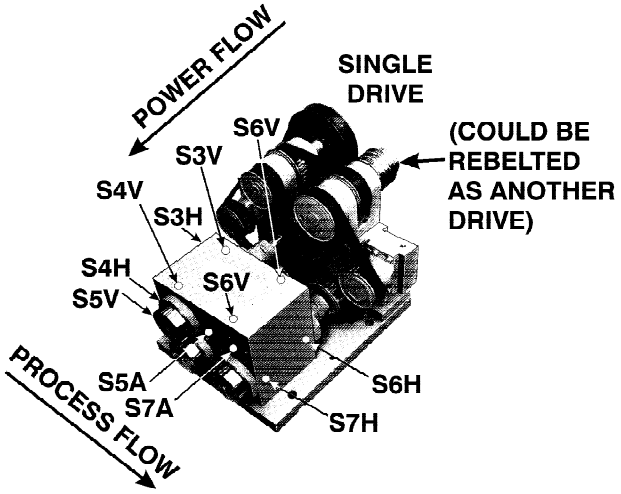


FIGURE 6
EXAMPLE OF CONSECUTIVE MEASUREMENT POINT LAYOUT ON PRECISION CARTRIDGE-TYPE SPRINDLES
(REF. Setco Cluster Spindles, Catalog #181D, Photo #3060, p.39)

Precision Multi-Gear Heads (Fig. 7) - Precision Multi-Gear Heads are typically driven by belts or couplings. They include parallel-axis spindles and worm-gear spindles in integral housings or external housings such as the bolt-on, sub-plate or slide-assembly design types. They may be used in, but not limitied to, cluster drilling, reaming, chamfering, spot-facing, countersinking and milling operations. It is recommended that measurement locations be numbered in the direction of power flow as well as in the direction of process flow as shown in Figure 7.

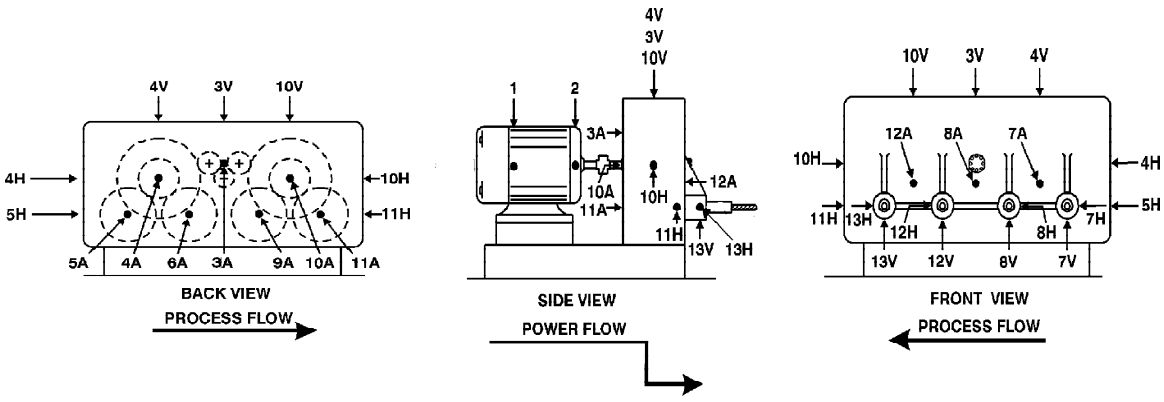


Figure 7.
EXAMPLE OF CONSECUTIVE MEASUREMENT POINT LAYOUT ON PRECISION MULTI-GEAR HEADS

© Copyright 2001 Technical Associates Of Charlotte, P.C.

Precision Grinding Wheel Heads (Fig. 8) - Precision Grinding Wheel Heads are usually belt-driven and include the center and centerless design types. They may consist of, but are not limited to internal cylindrical and external cylindrical, surface, chucking as well as tool and cutter grinding operations, in horizontal or vertical orientations.

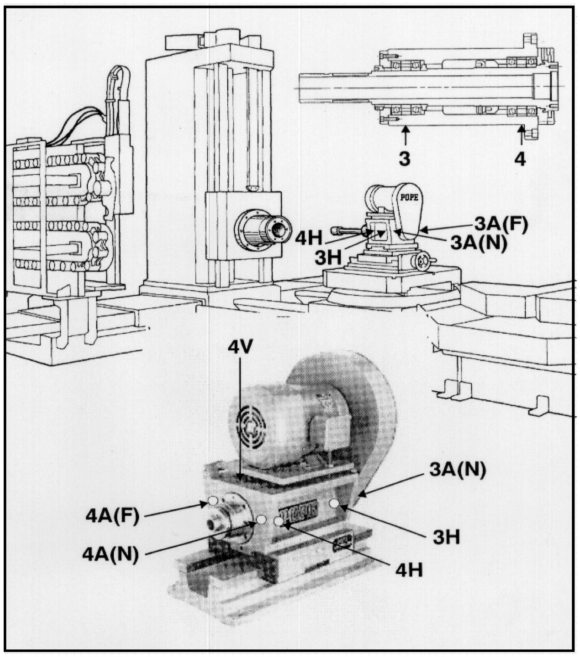


Figure 8.
EXAMPLE OF CONSECUTIVE MEASUREMENT
POINT LAYOUT ON PRECISION GRINDINGWHEEL HEADS
(REF. Pope Precision Spindles,SK-1418, Catalog No. 400, p.4)

Precision Integral-Motor Spindles (Fig. 9) - An example of a Precision Integral Motor-Spindle is shown in Figure 9. These units include the servo-design types. They may be used in, but not limited to drilling, reaming, chamfering, spotfacing, countersinking, boring, milling and internal deep-hole grinding operations. It is important that a cross-section view like that shown in Figure 9 is obtained for these spindles to ascertain the locations of bearings and to pinpoint good measurement locations relative to these bearings as shown in Figure 9.

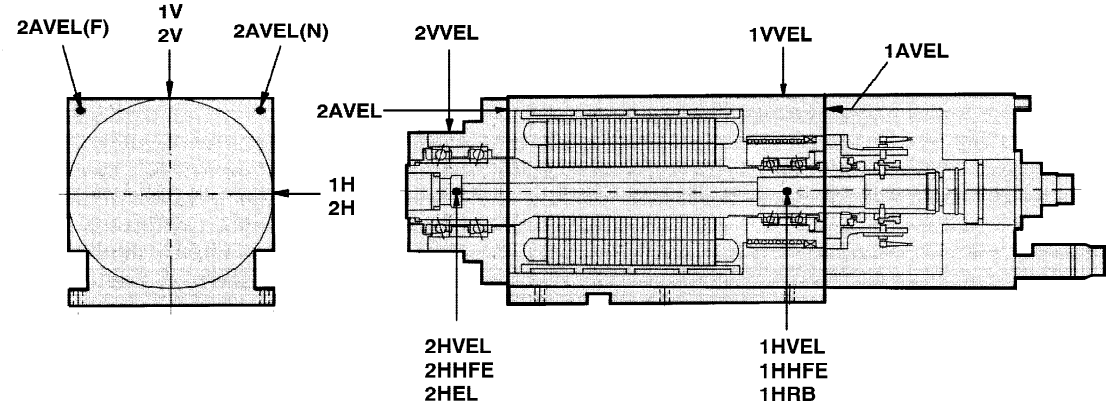


Figure 9.
EXAMPLE OF CONSECUTIVE MEASUREMENTPOINT LAYOUT
ON PRECISION INTEGRAL MOTOR SPINDLES
(REF. Setco Closed Loop Motorized Spindles, 230 Series, Catalog 995, p.10)

© Copyright 2001 Technical Associates Of Charlotte, P.C.

Vibration Alarms for Machine Tool Spindles:

Table II includes overall vibration alarm levels for machine tools. Note that the analyst must first determine the precision grade of the machine tool spindle in order to properly specify the overall alarm values for the spindle for which spectral band alarms are being specified ("Roughing Operations", "Machine Finishing", or "Critical Finishing" types). In addition, Section I on page 5 of Table III shows spectral band alarm measurement setups for such machine tool spindles based on the spindle type chosen. Note that alarm amplitudes for each of the six velocity bands are given in Table III and are expressed as a percentage of this overall alarm. Also, note that these alarm levels are broken down between bearing types (angular contact versus tapered roller bearings). For example, referring to Table III, note that alarm levels are slightly higher for tapered roller bearings than for angular contact bearings. When problems occur on bearings supporting these spindles, bearing fault frequencies and/or bearing natural frequencies are expected to be generated, particularly in Bands 5 and 6 of Case I. Higher alarm levels are specified for Bands 5 and 6 for tapered roller versus angular contact bearing since higher amplitude defect frequencies are typically generated by tapered roller bearings than those by angular contact bearings.

Table IIIB provides specific alarm levels at bearing fault frequencies (or harmonics) for each of the six machine tool types. These are separately specified for angular contact versus cylindrical and tapered roller bearings. For example, in the case of precision box spindles, Table IIIB shows that alarm levels of .005 in/sec should be applied for spindles supported by angular contact bearings versus .0075 in/sec for those supported by tapered roller bearings. Note that these specific alarm amplitudes apply at any bearing fault frequency or fault frequency harmonic (multiple) on any of the machine tool spindle types.

Table IIIC is also provided specifying maximum overall acceleration amplitudes for machine tool spindles (peak g's). These acceleration specifications all range between 600 and 600,000 CPM (10 - 10,000 Hz). Table IIIC specifies different acceleration alarm levels for angular contact versus tapered roller bearings; and in the case of tapered rollers, specifies higher acceleration levels for those operating above 1000 RPM (1.25 g) compared with those operating at or below 1000 RPM (0.75 g). This is due to the fact that higher acceleration amplitudes are generated by higher speed spindles supported by tapered roller bearings.

A note accompanying Table IIIC reminds analysts to take great care in mounting the transducer since measurements are acquired up to a rather high frequency of 600,000 CPM (10,000 Hz). Also, the analyst must ensure that the transducer itself is capable of taking accurate measurements up to this fairly high frequency.

7.151 EXAMPLES - Specification of Spectral Alarm Bands for Sample Machines:

Please refer to the Sample Machines shown in Figures 2 and 3, and then to the sample spectral alarm band tables which have been worked out for them. Note that the tables themselves are designed for direct input into the condition monitoring software once the alarm bands have been specified on them. Please note in particular how both the standard frequency and high frequency points are set up for the example with the 2-stage speed gear increaser for the gearbox shown in Figure 3.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

Metric Version Of Table II

TABLE II

TECHNICAL ASSOCIATES OF CHARLOTTE, P.C.

CRITERIA FOR OVERALL CONDITION RATING (RMS OVERALL VELOCITY, mm/sec.)*

1. Assuming Machine Speed = 600 to 60,000 RPM.
2. Assuming Measurements by Accelerometer or Velocity Pickup securely mounted as Close as Possible to Bearing Housing.
3. Assuming Machine Is Not Mounted on Vibration Isolators (for Isolated Machinery, Set Alarm 30% - 50% Higher).
4. Set Motor Alarms the Same as that for the Particular Machine Type unless Otherwise Noted.
5. Set Alarms on Individual External Gearbox Positions about 25% Higher than that for a particular Machine Type.

MACHINE TYPE	GOOD	FAIR	(WARNING) ALARM 1	(FAULT) ALARM 2
COOLING TOWER DRIVES				
Long, Hollow Drive Shaft	0 - 7.0	7.0 - 11.0	11.0	16.0
Close Coupled Belt Drive	0 - 5.0	5.0 - 7.5	7.5	12.0
Close Coupled Direct Drive	0 - 3.5	3.5 - 5.5	5.5	8.0
COMPRESSORS				
Reciprocating	0 - 6.0	6.0 - 9.0	9.0	13.5
Rotary Screw	0 - 5.5	5.5 - 8.0	8.0	12.0
Centrifugal With or W/O External Gearbox	0 - 3.5	3.5 - 5.5	5.5	8.0
Centrifugal - Integral Gear (Axial Meas.)	0 - 3.5	3.5 - 5.5	5.5	8.0
Centrifugal - Integral Gear (Radial Meas.)	0 - 2.5	2.5 - 4.5	4.5	6.5
BLOWERS (FANS)				
Lobe-Type Rotary	0 - 5.5	5.5 - 8.0	8.0	12.0
Belt-Driven Blowers	0 - 5.0	5.0 - 7.5	7.5	11.5
General Direct Drive Fans (with Coupling)	0 - 4.5	4.5 - 6.5	6.5	10.0
Primary Air Fans	0 - 4.5	4.5 - 6.5	6.5	10.0
Large Forced Draft Fans	0 - 3.5	3.5 - 5.5	5.5	8.0
Large Induced Draft Fans	0 - 3.0	3.0 - 5.0	5.0	7.0
Shaft-Mounted Integral Fan (Extended Motor Shaft)	0 - 3.0	3.0 - 5.0	5.0	7.0
Vane-Axial Fans	0 - 2.5	2.5 - 4.5	4.5	6.5
MOTOR/GENERATOR SETS				
Belt-Driven	0 - 5.0	5.0 - 7.5	7.5	12.0
Direct Coupled	0 - 3.5	3.5 - 5.5	5.5	8.0
CHILLERS				
Reciprocating	0 - 4.5	4.5 - 7.0	7.0	10.5
Centrifugal (Open-Air) - Motor & Comp. Separate	0 - 3.5	3.5 - 5.5	5.5	8.0
Centrifugal (Hermetic) - Motor & Impellers Inside	0 - 2.5	2.5 - 4.0	4.0	6.0
LARGE TURBINE/GENERATORS				
3600 RPM Turbine/Generators	0 - 3.0	3.0 - 5.0	5.0	7.0
1800 RPM Turbine/Generators	0 - 2.5	2.5 - 4.0	4.0	6.0
CENTRIFUGAL PUMPS				
Vertical Pumps (4m - 6m Height) } Height from Grade to Top Motor Bearing.	0 - 6.0	6.0 - 9.0	9.0	13.5
Vertical Pumps (3m - 4m Height) } May be Necessary to Spec. Lower Alarm	0 - 5.0	5.0 - 7.5	7.5	11.5
Vertical Pumps (2m - 3m Height) } for Lower Motor Bearing & for Upper	0 - 4.0	4.0 - 6.0	6.0	9.5
Vertical Pumps (0m -2m Height) } Pump Bearing (depending on height).	0 - 3.5	3.5 - 5.5	5.5	8.0
General Purpose Horizontal Pump - Direct Coupled	0 - 3.5	3.5 - 5.5	5.5	8.0
Boiler Feed Pumps - Horizontal Orientation	0 - 3.5	3.5 - 5.5	5.5	8.0
Hydraulic Pumps - Horizontal Orientation	0 - 2.0	2.0 - 3.5	3.5	5.5
MACHINE TOOLS				
Motor	0 - 1.2	1.2 - 1.8	1.8	2.7
Gearbox Input	0 - 1.8	1.8 - 2.7	2.7	4.0
Gearbox Output	0 - 1.2	1.2 - 1.8	1.8	2.7
Spindles:				
a. Roughing Operations	0 - .70	.70 - 1.2	1.2	1.8
b. Machine Finishing	0 - .45	.45 - .72	.72	1.1
c. Critical Finishing	0 - .27	.27 - .45	.45	.72

***NOTE:The "ALARM 1" and "ALARM 2" overall levels given above apply only to in-service machinery which has been operating for some time after initial installation and/or overhaul. They do not apply (and are not meant to serve as) Acceptance Criteria for either new or rebuilt machinery.**

COPYRIGHT© 1995 - TECHNICAL ASSOCIATES OF CHARLOTTE, P.C. R=1097-3RMSMM

© Copyright 2001 Technical Associates Of Charlotte, P.C.

TABLE III RECOMMENDED SPECTRAL ALARM BANDS FOR VARIOUS MACHINE TYPES (*See Notes Below Table)

CASE	ITEM	BAND 1	BAND 2	BAND 3	BAND 4	BAND 5	BAND 6
A. GENERAL ROLLING ELEMENT BEARING COMPONENTS WITHOUT VANES - Set F_{MAX} = 50X RPM (But, if Speed <1700 RPM or if Bearing is Tapered or Spherical Roller Type, See Notes b & c). (i.e., AC & DC Motors, Gearboxes) ***Gearboxes require second measurement with F_{MAX} and #Lines specified in Cases C and D .***							
	BAND LOWER FREQ.	1% F_{MAX}	1.2 X RPM	2.2 X RPM	3.2 X RPM	12.2 X RPM	50% F_{MAX}
	BAND HIGHER FREQ.	1.2 X RPM	2.2 X RPM	3.2 X RPM	12.2 X RPM	50% F_{MAX}	100% F_{MAX}
	BAND ALARM	90% OA ALARM	30% OA ALARM	25% OA ALARM	20% OA ALARM NTE 1.40 mm/sec	15% OA ALARM NTE 1.00 mm/sec	10% OA ALARM NTE 0.90 mm/sec
	DESCRIPTION OF BAND COVERAGE	(Subsynchronous thru 1X RPM)	(1.5 X - 2.0 X RPM)	(2.5 X - 3.0 X RPM)	(Fundamental Bearing Defect Frequencies)	(Lower Harmonic Bearing Frequencies)	(Higher Harmonic Bearing Freqs & Brg. Natl. Freqs.)
B. GENERAL SLEEVE BEARING MACHINES WITHOUT VANES (i.e., AC & DC Motors, Gearboxes) - Set F_{MAX} = 20X RPM. (For Gearboxes, also see Case C and D).							
	BAND LOWER FREQ.	0.2 X RPM	0.8 X RPM	1.8 X RPM	2.8 X RPM	3.8 X RPM	10.2 X RPM
	BAND HIGHER FREQ.	0.8 X RPM	1.8 X RPM	2.8 X RPM	3.8 X RPM	10.2 X RPM	100% F_{MAX}
	BAND ALARM	20% OA ALARM NTE 1.00 mm /sec	90% OA ALARM	30% OA ALARM	25% OA ALARM	20% OA ALARM	15% OA ALARM
	DESCRIPTION OF BAND COVERAGE	(Subsynchronous Band)	(1 X - 1.5 X RPM)	(2 X - 2.5 X RPM))	(3 X - 3.5 X RPM)	(4 X - 10 X RPM)	(10.5 X RPM - F_{MAX})
		Gearboxes require second measurement with F_{MAX} and #Lines specified in Cases C and D .					
C. GEARBOX SPECIAL HIGH FREQUENCY POINTS WITH KNOWN NUMBER OF TEETH - Set F_{MAX} = 3.25X Gear Mesh Frequency (GMF) - (However, see Case C Note Below)							
	BAND LOWER FREQ.	0.25 X GMF	0.75 X GMF	1.25 X GMF	1.75 X GMF	2.25 X GMF	2.75 X GMF
	BAND HIGHER FREQ.	0.75 X GMF	1.25 X GMF	1.75 X GMF	2.25 X GMF	2.75 X GMF	100% F_{MAX}
	BAND ALARM	25% OA ALARM	70% OA ALARM	25% OA ALARM	50% OA ALARM	25% OA ALARM	40% OA ALARM
	DESCRIPTION OF BAND COVERAGE	(0.5 X GMF & Sidebands)	(GMF & SIDEBANDS)	(1.5 X GMF & SIDEBANDS)	(2 X GMF & SIDEBANDS)	(2.5 X GMF & SIDEBANDS)	(3 X GMF & SIDEBANDS)
D. GEARBOX HIGH FREQUENCY POINTS WITH UNKNOWN NUMBER OF TEETH - Set F_{MAX} = 200X Shaft RPM at that particular location (However, see Notes a thru d)							
	BAND LOWER FREQ.	20 X RPM	50 X RPM	80 X RPM	110 X RPM	140 X RPM	170 X RPM
	BAND HIGHER FREQ.	50 X RPM	80 X RPM	110 X RPM	140 X RPM	170 X RPM	100% F_{MAX}
	BAND ALARM	60% OA ALARM	60% OA ALARM	50% OA ALARM	50% OA ALARM	40% OA ALARM	40% OA ALARM
	DESCRIPTION OF BAND COVERAGE	(20X - 50X RPM)	(50X - 80X RPM)	(80X - 110X RPM)	(110X - 140X RPM)	(140X - 170X RPM)	(170X RPM - F_{MAX})

NOTES: a. Obtain Overall Alarm Velocity from Table II entitled "TECHNICAL ASSOCIATES CRITERIA FOR CONDITION RATING" . Also 'F_{MAX}' = Maximum Frequency for Spectrum (CPM or Hz).
b. If bearing is a Tapered Roller Bearing or Spherical Roller Bearing, set F_{MAX} = 60 X RPM.
c. If Speed = 1400-1700 RPM, set F_{MAX} = 60 X RPM; if 1100-1399 RPM, set F_{MAX} = 80 X RPM; if 800-1099 RPM, set F_{MAX} = 100 X RPM; if 600-799 RPM, set F_{MAX} = 120XRPM.
(To Detect Bearing Natural Frequencies)

NOTE: On Gearboxes, specify these high frequency points checking up to 3.25 X GMF as well as points with normal frequency ranges (as in Case A or B above) on these same points (i.e., **2 separate measurements on same point with 2 different frequency ranges**). Specify F_{MAX} = 3.25 X GMF and locate points close to mesh locations of each set of gears using the specific gear mesh frequency at that particular location. However, remember that amplitudes at high frequencies approaching or exceeding the natural frequency of accelerometer mounting may be questionable (of transducer stud, magnet or probe mounting). In many cases, transducer may require stud or adhesive mounting and measurement of acceleration due to high F_{MAX}. **Use 1600-3200 FFT Lines** in order to allow detection of 1X RPM Sidebands of each meshing gear.

NOTE: a. Specify both high frequency as well as standard frequency measurement points at each location as in Case C. In many cases, transducer may require stud or adhesive mounting and measurement of acceleration due to high F_{MAX}.
b. **Use 3200 FFT Lines** to detect 1X RPM Sidebands of each meshing gear.
c. RPM = Shaft speed at the particular measurement point for which you are specifying spectral bands. If only input and output gearbox speeds are known in a multi-stage gearbox, assume an equal speed increment at each mesh (for example, for a 16:1 ratio, double reduction gearbox, assume approx. a 4.0 speed ratio at each of the mesh points as a starting point - refine when actual speeds are known).
d. When No. of Gear Teeth are found, replace these Spectral Alarm Bands **IMMEDIATELY** with setup shown in Case C above.

***NOTES:** 1. These Spectral Alarm Band Specs apply to **"ALARM 1"** for standard process and utility machinery such as centrifugal pumps, blowers, motors, FD & ID fans, motor/generator sets, refrigeration chillers, centrifugal air compressors, vacuum pumps, boiler feed pumps, gearboxes, etc. **"ALARM 2"** (higher alarm band setting) should have alarm levels 50% greater than **"ALARM 1"**.
2. These Specs do not apply to other machine types such as reciprocating, axial screw or lobe-type rotary compressors; diesel engines; gas turbines; large turbine/generators; exciters; rotary blowers; pulverizers, etc. Spectral Bands for these types normally have to be 'custom-designed' for each of these particular machines or group of similar machines, often after complete vibration spectra have been captured one or more times on these machines.
3. "NTE" = Not to Exceed; "OA ALARM" = Overall Alarm Specified in Table II.
4. **Frequency Resolution = 800 Lines** unless otherwise instructed; **If F_{MAX} > 100X RPM**, increase resolution to **1600 lines** unless otherwise instructed; **Use 50% Overlap Averaging.**

© COPYRIGHT 2000

© Copyright 2001 Technical Associates Of Charlotte, P.C.

TABLE III

RECOMMENDED SPECTRAL ALARM BANDS FOR VARIOUS MACHINE TYPES (*See Notes Below Table)

CASE	ITEM	BAND 1	BAND 2	BAND 3	BAND 4	BAND 5	BAND 6
E. AC INDUCTION MOTOR ELECTRICAL ROTOR BAR PASS FREQUENCY POINT (MEASURE ON MOTOR OUTBOARD BEARING HORIZONTAL) - Set F_{MAX} = 360,000 CPM (See Notes a thru d).							
	BAND LOWER FREQ.	30,000 CPM	85,000 CPM	140,000 CPM	195,000 CPM	250,000 CPM	305,000 CPM
	BAND HIGHER FREQ.	85,000 CPM	140,000 CPM	195,000 CPM	250,000 CPM	305,000 CPM	360,000 CPM
	BAND ALARM	1.00 mm/sec.	1.00 mm/sec.	0.90 mm/sec.	0.70 mm/sec	0.55 mm/sec	0.55 mm/sec
	DESCRIPTION OF BAND COVERAGE	(30K - 85K)	(85K - 140K)	(140K - 195K)	(195K - 250K)	(250K - 305K)	(305K - 360K)
NOTES: a. Purpose of this Point on each motor is to detect possible 1X and 2X Rotor Bar Pass Frequencies (RBPF) which may be accompanied by 2X line frequency sidebands (normally at 7200CPM). b. Case E applies to motors ranging in speeds from nominal 900 to 3600 RPM. c. Ensure that transducer is well mounted on the motor housing due to the rather high F_{MAX}. Normally, it should be taken in the horizontal direction on the outboard motor bearing housing. d. Should use **1600 FFT Lines** & 8 Averages to clearly identify possible Rotor Bar Pass Frequencies as well as 2X Line Frequency and/or 1X RPM Sidebands.							
F. AC INDUCTION MOTOR ELECTRICAL MEASUREMENT POINT ON MOTOR INBOARD BEARING HORIZONTAL - Set F_{MAX} = 24,000 CPM (For nominal 900 - 3600 RPM motors).							
	BAND LOWER FREQ.	1.8 X Line Freq.	3.8 X Line Freq.	NOT REQUIRED	NOT REQUIRED	NOT REQUIRED	NOT REQUIRED
	BAND HIGHER FREQ.	2.2 X Line Freq.	4.2 X Line Freq.				
	BAND ALARM	1.25 mm/sec	0.60 mm/sec				
	DESCRIPTION OF BAND COVERAGE	(2 F_{LINE} ± Pole Pass Freq. Sidebands)	(4 F_{LINE} ± Pole Pass Freq. Sidebands)				
NOTES: a. Purpose of this Point is to attempt to separate Mechanical and Electrical vibration peaks, particularly in vicinity of 2X Line Frequency and 4X Line Frequency. b. Should use **6400 FFT Lines** and 2 Averages to clearly separate Mechanical & Electrical Frequencies and to detect Pole Pass Freq. Sidebands (F_P = Slip Freq. X #Poles). c. This measurement is **Required** on Each Motor to Detect Pole Pass Freq. Sidebands surrounding either 1X RPM Harmonics and/or 2X Line Frequency.							
G. CENTRIFUGAL COMPRESSORS, BLOWERS AND PUMPS **TYPE 1. DRIVEN CENTRIFUGAL COMPONENT WITH KNOWN NUMBER OF VANES (OR BLADES) AND ROLLING ELEMENT BEARINGS - Set F_{MAX} = 50X RPM (See Notes a thru c).**							
	BAND LOWER FREQ.	0.4 X RPM	1.2 X RPM	2.2 X RPM (BPF - 1.2 X RPM) 35% OA ALARM NTE 2.15 mm/sec. (2.5 X - Funda. Bearing Defect Frequencies)	BPF - 1.2 X RPM BPF + 1.2 X RPM 60% OA ALARM NTE 4.30 mm/sec. (BPF ± 1 X Sidebands) BPF = Blade Pass Freq.	BPF + 1.2 X RPM 50% F_{MAX} 35% OA ALARM NTE 2.15 mm/sec. (Lower Harmonic Bearing Freqs. & BPF Harmonics)	50% F_{MAX} 100% F_{MAX} 20% OA ALARM NTE 1.30 mm/sec. (Higher Harmonic Bearing Freqs. & Brg. Natl. Freqs.)
	BAND HIGHER FREQ.	1.2 X RPM	2.2 X RPM				
	BAND ALARM	90% OA ALARM	30% OA ALARM				
	DESCRIPTION OF BAND COVERAGE	(Subsynchronous thru 1 X RPM)	(1.5 -2.0 X RPM)				
NOTES: a. Specify separate set of spectral alarm bands for Driver (as shown in Case A or B for motor, gearbox, etc.) where BPF is not involved. b. "BPF" = Blade Pass Frequency = No. of Blades (or Vanes) X RPM. If bearing is a Tapered Roller Bearing or Spherical Roller Bearing, Set F_{MAX} =60X RPM unless speed < 1700 RPM (see note b in Type 1). c. If Speed = 1400-1700 RPM, set F_{MAX} = 50X RPM; if 1100-1399 RPM, set F_{MAX} = 80X RPM; if 800-1099 RPM, set F_{MAX} = 100 X RPM④; if 600-799 RPM, set F_{MAX} = 120X RPM④.							
TYPE 2A. CENTRIFUGAL PUMPS WITH UNKNOWN NUMBER OF VANES AND ROLLING ELEMENT BEARINGS -Set F_{MAX} = 50X RPM (See Notes a thru c).							
	BAND LOWER FREQ.	1% F_{MAX}	1.2 X RPM	2.2 X RPM	3.2 X RPM	6.8 X RPM	50% F_{MAX}
	BAND HIGHER FREQ.	1.2 X RPM	2.2 X RPM	3.2 X RPM	6.8 X RPM	50% F_{MAX}	100% F_{MAX}
	BAND ALARM	90% OA ALARM	30% OA ALARM	25% OA ALARM	60% OA ALARM NTE 3.30 mm/sec.	30% OA ALARM NTE 2.15 mm/sec.	20% OA ALARM NTE 1.30 mm/sec.
	DESCRIPTION OF BAND COVERAGE	(Subsynchronous thru 1 X RPM)	(1.5 -2.0 X RPM)	(2.5-3.0 X RPM)	(Possible BPF for Pumps)	(Lower Harmonic Bearing Freq. & BPF Harmonics)	(Higher Harmonic Bearing Freqs. & Brg. Natl. Freqs.)
NOTES: a. If bearing is a Tapered roller Bearing or Spherical Roller Bearing, Set F_{MAX} = 60 X RPM unless Speed < 1700 RPM. b. If Speed = 1400-1700 RPM, set F_{MAX} = 60X RPM; if 1100-1399 RPM, set F_{MAX} = 80X RPM; if 800-1099 RPM, set F_{MAX} = 100 X RPM④; if 600-799 RPM, set F_{MAX} = 120X RPM④. c. When Number of Vanes is confirmed, replace these Spectral Alarm Bands **IMMEDIATELY** with setup shown in Case G. Type 1.							

*NOTE: 1. These Spectral Alarm Band Specs apply to **"ALARM 1"** for standard process and utility machinery such as centrifugal pumps, blowers, motors, FD & ID fans, motor/generator sets refrigeration chillers, centrifugal air compressors, vacuum pumps, boiler feed pumps, gearboxes, etc. **"ALARM 2"** (higher alarm band setting) should have alarm levels 50% greater than **"ALARM 1"**.
2. These Specs do not apply to other machine types such as reciprocating, axial screw or lobe-type rotary compressors; diesel engines; gas turbines; large turbine/generators; exciters; rotary blowers; pulverizers, etc. Spectral Bands for these types normally have to be "custom-designed" for each of these particular machines or group of similar machines, often after complete vibration spectra have been captured one or more times on these machines.
3. "NTE" = Not to Exceed; "OA ALARM" = Overall Alarm Specified in Table II.
4. **Frequency Resolution = 800 Lines** unless otherwise instructed; **If F_{MAX} > 100X RPM**, increase resolution to **1600 lines** unless otherwise instructed; **Use 50% Overlap Averaging**.

© COPYRIGHT 2000

© Copyright 2001 Technical Associates Of Charlotte, P.C.

TABLE III

RECOMMENDED SPECTRAL ALARM BANDS FOR VARIOUS MACHINE TYPES (*See Notes Below Table)

CASE	ITEM	BAND 1	BAND 2	BAND 3	BAND 4	BAND 5	BAND 6
TYPE 2B. CENTRIFUGAL BLOWERS & COMPRESSORS WITH UNKNOWN NUMBER OF BLADES AND ROLLING ELEMENT BEARINGS - Set F_{MAX} = 50X RPM (See Notes a thru c).							
	BAND LOWER FREQ. BAND HIGHER FREQ. BAND ALARM	1% F_{MAX} 1.2 X RPM 90% OA ALARM	1.2 X RPM 3.2 X RPM 35% OA ALARM	3.2 X RPM 7.8 X RPM 30% OA ALARM	7.8 X RPM 12.8 X RPM 40% OA ALARM NTE 1.80 mm/sec.	12.8 X RPM 50% F_{MAX} 20% OA ALARM NTE 1.40 mm/sec	50% F_{MAX} 100% F_{MAX} 15% OA ALARM NTE 1.00 mm/sec
	DESCRIPTION OF BAND COVERAGE	(Subsynchronous thru 1X RPM)	(1.5 - 3.0X RPM)	(3.5 - 7.5 X RPM)	(Possible BPF for Fans & Compressors)	(Lower Harmonic Bearing Freq. & BPF Harmonics)	(Higher Harmonic Beaing Freq. & BPF Harmonics)
NOTES: a. If bearing is a Tapered Roller Bearing or Spherical Roller Bearing, Set F_{MAX} = 60 X RPM unless Speed < 1700 RPM. (see Note b). b. If Speed = 1400-1700 RPM, set F_{MAX} = 60X RPM; if 1100-1399 RPM, set F_{MAX} = 80X RPM; if 800-1099 RPM, set F_{MAX} = 100 X RPM; if 600-799 RPM, set F_{MAX} = 120X RPM. c. When Number of Blades is confirmed, replace these Spectral Alarm Bands **IMMEDIATELY** with setup shown in Case G, Type 1.							
TYPE 3 . DRIVEN CENTRIFUGAL COMPONENT WITH KNOWN NUMBER OF VANES (OR BLADES) AND WITH SLEEVE BEARINGS - Set F_{MAX} =20X RPM or 1.2X BPF, Whichever is Greater .							
	BAND LOWER FREQ. BAND HIGHER FREQ. BAND ALARM	0.3 X RPM 0.8 X RPM 20% OA ALARM NTE 1.00 mm/sec	0.8 X RPM 1.8 X RPM 90% OA ALARM	1.8 X RPM 3.8 X RPM 40% OA ALARM	3.8 X RPM (BPF - 1.2 X RPM) 30% OA ALARM	(BPF - 1.2 X RPM) (BPF + 1.2 X RPM) 70% OA ALARM NTE 4.30 mm/sec	(BPF + 1.2 X RPM) 100% F_{MAX} 30% OA ALARM NTE 1.80 mm/sec
	DESCRIPTION OF BAND COVERAGE	(Subsynchroncus Band)	(1X - 1.5X RPM)	(2X - 3.5 X RPM)	(4X - Lower RPM Harmonics)	(BPF ± 1X Sidebands)	(Higher RPM Harmonics and BPF Harmonics)
NOTES: a. Specify separate set of spectral alarm bands for Driver (as shown in Case A or B for motor, gearbox, etc.) where BPF is not involved. b. "BPF" = Blade Pass Frequency = No. of Blades (or Vanes) X RPM. c. If Number of Vanes >25, set Band 1 Lower Frequency = 0.4 X RPM.							
TYPE 4A. CENTRIFUGAL, PUMPS EQUIPPED WITH UNKNOWN NUMBER OF VANES AND WITH SLEEVE BEARINGS - Set F_{MAX} = 20X RPM (See Notes a and b).							
	BAND LOWER FREQ. BAND HIGHER FREQ. BAND ALARM	0.3 X RPM 0.8 X RPM 20% OA ALARM NTE 1.00 mm/sec.	0.8 X RPM 1.8 X RPM 90% OA ALARM	1.8 X RPM 3.8 X RPM 40% OA ALARM	3.8 X RPM 7.2 X RPM 70% OA ALARM NTE 4.30 mm/sec.	7.2 X RPM 9.8 X RPM 25% OA ALARM	9.8 X RPM 100% F_{MAX} 35% OA ALARM NTE 2.15 mm/sec.
	DESCRIPTION OF BAND COVERAGE	(Subsynchronous Band)	(1X - 1.5X RPM)	(2X - 3.5X RPM)	(Possible BPF for Pumps)	(8X - 9.5X RPM)	(10X RPM - F_{MAX})
NOTES: a. Specify separate set of spectral alarm bands for Driver (as shown in Case A or B for motor, gearbox, etc.) where BPF is not involved. b. When Number of Vanes is found, replace these Spectral Alarm Bands **IMMEDIATELY** with setup shown in Case G, Type 3.							
TYPE 4B. CENTRIFUGAL BLOWERS & COMPRESSORS WITH UNKNOWN NUMBER OF BLADES AND WITH SLEEVE BEARINGS -Set F_{MAX} = 30X RPM (See Notes a and b).							
	BAND LOWER FREQ. BAND HIGHER FREQ. BAND ALARM	0.3 X RPM 0.8 X RPM 20% OA ALARM NTE 1.00 mm/sec.	0.8 X RPM 1.8 X RPM 90% OA ALARM	1.8 X RPM 3.2 X RPM 40% OA ALARM	3.2 X RPM 7.2 X RPM 30% OA ALARM	7.2 X RPM 12.8 X RPM 40% OA ALARM NTE 1.80 mm/sec.	12.8 X RPM 100% F_{MAX} 20% OA ALARM NTE 1.00 mm/sec.
	DESCRIPTION OF BAND COVERAGE	(Subsynchronous Band)	(1X - 1.5X RPM)	(2X - 3X RPM)	(3.5 - 7.5X RPM)	(Possible BPF for Fans and Compressors)	(13X RPM - F_{MAX})
NOTES: a. Specify separate set of spectral alarm bands for Driver (as shown in Case A or B for motor, gearbox, etc.) where BPF is not involved. b. When Number of Blades is found, replace these Spectral Alarm Bands **IMMEDIATELY** with setup shown in Case G, Type 3.							

*NOTE: 1. These Spectral Alarm Band Specs apply to **"ALARM 1"** for standard process and utility machinery such as centrifugal pumps, blowers, motors, FD & ID fans, motor/generator sets, refrigeration chillers, centrifugal air compressors, vacuum pumps, boiler feed pumps, gearboxes, etc. **"ALARM 2"** (higher alarm band setting) should have alarm levels 50% greater than **"ALARM 1"**.

2. These Specs do not apply to other machine types such as reciprocating, axial screw or lobe-type rotary compressors; diesel engines; gas turbines; large turbine/generators; exciters; rotary blowers; pulverizers, etc. Spectral Bands for these types normally have to be "custom-designed" for each of these particular machines or group of similar machines, often after complete vibration spectra have been captured one or more times on these machines.

3. "NTE" = Not to Exceed; "OA ALARM" = Overall Alarm Specified in Table II.

4. **Frequency Resolution = 800 Lines** unless otherwise instructed; **If F_{MAX} > 100X RPM**, increase resolution to **1600 lines** unless otherwise instructed. **Use 50% Overlap Averaging.**

© COPYRIGHT 2000

© Copyright 2001 Technical Associates Of Charlotte, P.C.

TABLE III

RECOMMENDED SPECTRAL ALARM BANDS FOR VARIOUS MACHINE TYPES (*See Notes Below Table)

CASE	ITEM	BAND 1	BAND 2	BAND 3	BAND 4	BAND 5	BAND 6
H. DC MOTORS: FULL-WAVE AND HALF-WAVE SCR CONTROLLED RECTIFIER (Special Measurements in Addition to Standard Spectral Setups in Cases A & B).							
TYPE 1. FULL-WAVE RECTIFIED MOTORS (MEASURE ON COMMUTATOR-END BEARING HORIZONTAL)- Set F_{MAX} = 24,000 CPM (See Notes a thru c).							
	BAND LOWER FREQ.	3300 CPM	6900 CPM	10,500 CPM	14,100 CPM	17,700 CPM	21,600 - 1.2 X RPM
	BAND HIGHER FREQ.	3900 CPM	7500 CPM	11,100 CPM	14,700 CPM	18,300 CPM	21,600 + 1.2 X RPM
	BAND ALARM	0.35 mm/sec	0.35 mm/sec	0.35 mm/sec	0.35 mm/sec	0.35 mm/sec	1.45 mm/sec
	DESCRIPTION OF BAND COVERAGE	(1 X Line Freq., F_L)	(2 X Line Freq.)	(3 X Line Freq.)	(4 X Line Freq.)	(5 X Line Freq.)	(SCR Firing Freq. ± 1 X RPM Sidebands)
NOTES: a. Purpose of this Special Measurement Point is to Detect Problems with Armature Windings, Commutators, SCR's, Firing Cards, Comparitor Cards, Fuses, etc. b. Must use 6400 Lines and 2 Averages to Detect 1 X RPM and 1 X RPM Sidebands around the SCR Firing Frequency; Use 50% Overlap Averaging. c. Assumes Line Frequency = 60 Hz (3600 CPM); Adjust Frequency Span of Each Band if Line Freq. Differs from 60 Hz.							
TYPE 2 . HALF-WAVE RECTIFIED MOTORS (MEASURE ON COMMUTATOR-END BEARING HORIZONTAL) - Set F_{MAX} =24,000 CPM (See Notes a thru c).							
	BAND LOWER FREQ.	3300 CPM	6900 CPM	10,800 - 1.2 X RPM	14,100 CPM	17,700 CPM	21,600 - 1.2 X RPM
	BAND HIGHER FREQ.	3900 CPM	7500 CPM	10,800 + 1.2 X RPM	14,700 CPM	18,300 CPM	21,600 + 1.2 X RPM
	BAND ALARM	0.35 mm/sec	0.35 mm/sec	1.45 mm/sec	0.30 mm/sec	0.30 mm/sec	0.70 mm/sec
	DESCRIPTION OF BAND COVERAGE	(1 X Line Freq., F_L)	(2 X Line Freq.)	(SCR Firing Freq. ± 1 X RPM Sidebands)	(4 X Line Freq.)	(5 X Line Freq.)	(2 X SCR Firing Freq. ± 1 X RPM Sidebands)
NOTES: a. Purpose of this Special Measurement Point is to Detect Problems with Armature Windings, Commutators, SCR's, Firing Cards, Comparitor Cards, Fuses, etc. b. Must use 6400 Lines and 2 Averages to Detect 1 X RPM and 1 X RPM Sidebands around the SCR Firing Frequency; Use 50% Overlap Averaging. c. Assumes Line Frequency = 60 Hz (3600 CPM); Adjust Frequency Span of Each Band if Line Freq. Differs from 60 Hz.							
TYPE 3. SPECIAL POINT FOR ELECTRICALLY - INDUCED FLUTING ON FULL-WAVE OR HALF-WAVE RECTIFIED DC MOTORS - SET F_{MAX} = 180,000 CPM (See Notes a thru e).							
	BAND LOWER FREQ.	(SCR - 2.2 X RPM)	(2 X SCR - 2.2 X RPM)	(3 X SCR - 2.2 X RPM)	(4 X SCR - 2.2 X RPM)	(5 X SCR - 2.2 X RPM)	120,000 CPM
	BAND HIGHER FREQ.	(SCR + 2.2 X RPM)	(2 X SCR + 2.2 X RPM)	(3 X SCR + 2.2 X RPM)	(4 X SCR + 2.2 X RPM)	120,000 CPM	180,000 CPM
	BAND ALARM	1.45 mm/sec	0.70 mm/sec	0.45 mm/sec	0.40 mm/sec	0.30 mm/sec	0.30 mm/sec
	DESCRIPTION OF BAND COVERAGE	(SCR Firing Freq. ± 2 X RPM Sidebands)	(2 X SCR Firing Freq. ± 2 X RPM Sidebands)	(3 X SCR Firing Freq. ± 2 X RPM Sidebands)	(4 X SCR Firing Freq. ± 2 X RPM Sidebands)	(BPFO or BPFI Sidebands Around Carrier Freq.)	(BPFO or BPFI Sidebands Around Carrier Freq.)
NOTES: a. Primary Purpose of this Special Measurement is to Detect Fluting normally indicated by BPFO and/or BPFI Sidebands around Carrier Frequency which Ranges from 100,000 to 150,000 CPM. b. SCR Firing Freq. for Half-Wave Rectified DC Motor = 10,800 CPM (180 Hz); but 21,600 CPM (360 Hz) for Full-Wave Rectified DC Motor (with a line frequency (F_L) of 60Hz). c. Must use 3200 lines & 8 averages with F_{MAX} of 180,000 CPM in order to detect electrically induced fluting which is indicated by Spectrum Described in Note a. d. Should take this measurement on Both the Outboard and Inboard Motor Bearings since fluting can occur on either bearing. Use 50% Overlap Averaging. e. Take data in Horizontal Direction on Both bearings and ensure tranducer is well mounted due to the rather high F_{MAX}.							

*NOTE: 1. These Spectral Alarm Band Specs apply to **"ALARM 1"** for standard process and utility machinery such as centrifugal pumps, blowers, motors, FD & ID fans, motor/generator sets, refrigeration chillers, centrifugal air compressors, vacuum pumps, boiler feed pumps, gearboxes, etc. **"ALARM 2"** (higher alarm band setting) should have alarm levels 50% greater than **"ALARM 1"**.

2. These Specs do not apply to other machine types such as reciprocating, axial screw or lobe-type rotary compressors; diesel engines; gas turbines; large turbine/generators; exciters; rotary blowers; pulverizers, etc. Spectral Bands for these types normally have to be "custom-designed" for each of these particular machines or group of similar machines, often after complete vibration spectra have been captured one or more times on these machines.

3. "NTE" = Not to Exceed; "OA ALARM" = Overall Alarm Specified in Table II.

4. **Frequency Resolution = 800 Lines** unless otherwise instructed; **If F_{MAX} > 100X RPM**, increase resolution to **1600 lines** unless otherwise instructed; **Use 50% Overlap Averaging.**

© COPYRIGHT 2000

© Copyright 2001 Technical Associates Of Charlotte, P.C.

TABLE III

RECOMMENDED SPECTRAL ALARM BANDS FOR VARIOUS MACHINE TYPES (*See Notes Below Table)

CASE	ITEM	BAND 1	BAND 2	BAND 3	BAND 4	BAND 5	BAND 6
I. MACHINE TOOL SPINDLES (see Table II and Text Section I) - Set F_{MAX} = 50X RPM (But, if speed < 1700 RPM or if Bearing is Tapered or Spherical Roller Type, See Notes a thru c).							
	BAND LOWER FREQ.	0.3 X RPM	0.8 X RPM	1.2 X RPM	2.2 X RPM	3.8 X RPM	50% F_{MAX}
	BAND HIGHER FREQ.	0.8 X RPM	1.2 X RPM	2.2 X RPM	3.8 X RPM	50% F_{MAX}	100% F_{MAX}
	BAND ALARMS:						
	Angular Contact	30% OA ALARM	60% OA ALARM	40% OA ALARM	40% OA ALARM	20% OA ALARM	10% OA ALARM
	Tapered Roller	30% OA ALARM	60% OA ALARM	40% OA ALARM	40% OA ALARM	30% OA ALARM	15% OA ALARM
	DESCRIPTION OF BAND COVERAGE	(FTF & Subsynch.)	(1X RPM)	(1.5X - 2.0X RPM)	(3X RPM & BSF)	(BSF, BPFO, BPFI & Harmonics)	(Bearing/Housing Natural Freqs.)

NOTES: a. Obtain Overall Alarm Velocity from Table II entitled "TECHNICAL ASSOCIATES CRITERIA FOR CONDITION RATING". Also "F_{MAX}" = Maximum Frequency for Spectrum (CPM or Hz).
b. If bearing is a Tapered Roller Bearing or Spherical Roller Bearing, set F_{MAX} = 60 X RPM.
c. If Speed = 1400-1700 RPM, set F_{MAX} = 60 X RPM; if 1100-1399 RPM, set F_{MAX} = 80 X RPM; if 800-1099 RPM, set F_{MAX} = 100 X RPM; if 600-799 RPM, set F_{MAX} = 120XRPM.
(To Detect Bearing Natural Frequencies) Gearboxes require second measurement with F_{MAX} and #Lines specified in Case C and D.
D. See Table IIIB which specifies maximum allowable amplitudes for discrete

TABLE IIIB. MAXIMUM ALLOWABLE AMPLITUDE AT BPFO, BPFI, BSF OR ANY BEARING FREQUENCY HARMONIC (Multiple)ON MACHINE TOOL SPINDLES (See Note below)

SPINDLE TYPE	TEXT FIG. NO.	MAX. BEARING FREQ. AMPLITUDE	
		ANGULAR CONTACT	CYLIND./ TAPERED
Precision Box Spindle	Fig. 2	0.10 mm/sec	0.13 mm/sec
Precision Cartridge-Type Spindle	Fig. 3	0.10 mm/sec	0.13 mm/sec
Precision Multi-Gear Head Spindle	Fig. 4	0.10 mm/sec	N/A
Precision Multi-Gear Head Gearbox	Fig. 4	0.13 mm/sec	0.13 mm/sec
Precision Grinding Wheel Head	Fig. 5	0.10 mm/sec	0.13 mm/sec
Precision Integral-Motor Spindle	Fig. 6	0.10 mm/sec	0.10 mm/sec

NOTE: Information in Case I is derived from Technical Associates' ***"Vibration Acceptance Criteria for New and Rebuilt Machinery"*** (Ref. 40)

TABLE IIIC. MAXIMUM BAND LIMITED OVERALL PEAK ACCELERATION FOR MACHINE TOOL SPINDLES (See Note below)

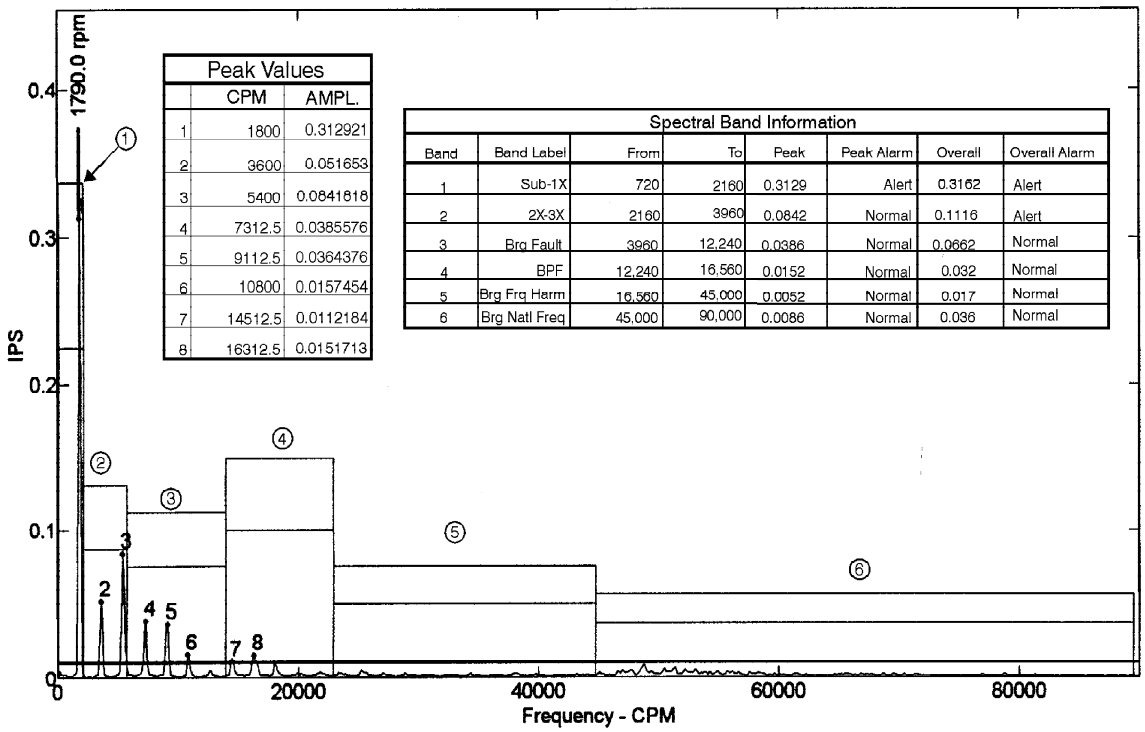
BEARING TYPE	SPINDLE SPEED	FREQUENCY BAND*.	PEAK ACCEL.
Angular Contact	Any Speed	600 - 600,000 CPM	0.50g
Tapered Roller	< 1000 RPM	600 - 600,000 CPM	0.75g
Tapered Roller	> 1000 RPM	600 - 600,000 CPM	1.25g

*NOTE: Great care must be taken in using a proper transducer and in mounting the transducer since measurements are required up to these high frequencies.

*NOTE: 1. These Spectral Alarm Band Specs apply to **"ALARM 1"** for standard process and utility machinery such as centrifugal pumps, blowers, motors, FD & ID fans,motor/generator sets, refrigeration chillers, centrifugal air compressors, vacuum pumps, boiler feed pumps, gearboxes, etc. **"ALARM 2"** (higher alarm band setting) should have alarm levels 50% greater than **"ALARM 1"**
2. These Specs do not apply to other machine types such as reciprocating, axial screw or lobe-type rotary compressors; diesel engines; gas turbines; large turbine/generators; exciters; rotary blowers; pulverizers, etc. Spectral Bands for these types normally have to be "custom-designed" for each of these particular machines or group of similar machines, often after complete vibration spectra have been captured one or more times on these machines.
3. 'NTE' = Not to Exceed; "OA ALARM" = Overall Alarm Specified in Table II.
4. Frequency Resolution = 800 Lines unless otherwise instructed. If F_{MAX} > 10CX RPM, increase resolution to 1600 lines unless otherwise instructed. Use 50% overlap Averaging.

© COPYRIGHT 2000

FIGURE 10
SPECTRAL ALARM BAND SPECS ON A HORIZONTAL PUMP (8 Vanes)



Peak Values		
	CPM	AMPL.
1	1800	0.312921
2	3600	0.051653
3	5400	0.0841818
4	7312.5	0.0385576
5	9112.5	0.0364376
6	10800	0.0157454
7	14512.5	0.0112184
8	16312.5	0.0151713

Spectral Band Information							
Band	Band Label	From	To	Peak	Peak Alarm	Overall	Overall Alarm
1	Sub-1X	720	2160	0.3129	Alert	0.3162	Alert
2	2X-3X	2160	3960	0.0842	Normal	0.1116	Alert
3	Brg Fault	3960	12,240	0.0386	Normal	0.0662	Normal
4	BPF	12,240	16,560	0.0152	Normal	0.032	Normal
5	Brg Frq Harm	16,560	45,000	0.0052	Normal	0.017	Normal
6	Brg Natl Freq	45,000	90,000	0.0086	Normal	0.036	Normal

EXAMPLE 1 TECHNICAL ASSOCIATES OF CHARLOTTE, P.C. (704) 333-9011

CLIENT NAME: XYZ COMPANY
MACHINE NAME: HORIZ. PUMP 2

BY: JEB
DATE: 5/12/2000

SPECTRAL ALARM BAND COMPUTER ENTRY FORM

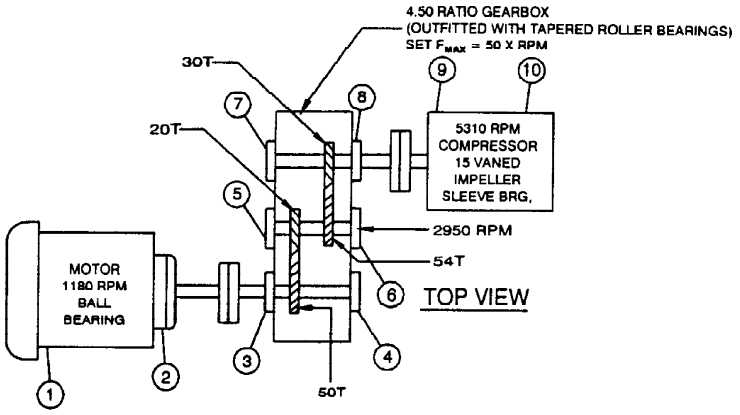
POSITION	UNITS	ALARM	RPM	BRGS.	F_{MAX}	#LINES	PARAMETER	BAND 1	BAND 2	BAND 3	BAND 4	BAND 5	BAND 6
1A, 1H, 1V 2A, 2H, 2V **(Motor Points As Per Case A)**	in/sec	.300	1780	Ball	90K	800	LOWER FREQ. HIGHER FREQ. BAND ALARM	900 2160 .270	2160 3960 .090	3960 5760 .075	5760 21,960 .060	21,960 45,000 .045	45,000 90,000 .030
1 BAR **(Rotor Bar Pass Freq. Point From Case E)**	in/sec	.300	1780	Ball	360K	1600	LOWER FREQ. HIGHER FREQ. BAND ALARM	30,000 85,000 .060	85,000 140,000 .055	140,000 195,000 .050	195,000 250,000 .040	250,000 305,000 .035	305,000 360,000 .030
2 EL **(Motor Points As Per Case A)**	in/sec	.300	1780	Ball	24K	6400	LOWER FREQ. HIGHER FREQ. BAND ALARM	6480 7920 .070	13,680 15,120 .035	N/A	N/A	N/A	N/A
1H, 2H, 3H, 4H	gSE	.700	1780	Ball	------	N/A	LOWER FREQ. HIGHER FREQ. BAND ALARM	NO ALARMS BANDS FOR HIGH FREQ. ENVEL. SPECTRA					
3A, 3H, 3V 4A, 4H, 4V **(Pump Points As Per Case G, Type 1 with 8 Vanes)**	in/sec	.300	1780	Ball	90K	800	LOWER FREQ. HIGHER FREQ. BAND ALARM	720 2160 .270	2160 3960 .090	3960 12,240 .105	12,240 16,560 .180	16,560 45,000 .105	45,000 90,000 .060

OTHER MACHINES WHICH WILL HAVE DUPLICATE SPECTRAL ALARM BANDS:				
HORIZ PUMP 3	HORIZ PUMP 12			
HORIZ PUMP 5	HORIZ PUMP 16			
HORIZ PUMP 7				
HORIZ PUMP 10				

© Copyright 2001 Technical Associates Of Charlotte, P.C.

Metric Version Of Example 2

FIGURE 3
EXAMPLE #2- COMPRESSOR DRIVEN BY 2-STAGE GEAR BOX



GMF_1 = (50T)(1180 RPM) = (20T)(2950 RPM)
GMF_1 = 59,000 CPM
POS. 3HI F_{MAX} = GMF X 3.25
= 191,750 CPM
USE POS 3HI F_{MAX} = 200,00 CPM

GMF_2 = 54T X 2950 RPM = (30T)(5310 RPM)
GMF_2 = 159,300 CPM
POS. 6HI F_{MAX} = 3.25 X 159,300
= 517,725 CPM
USE POS 6HI F_{MAX} = 540,00 CPM

(Must Measure Acceleration Here)

EXAMPLE 2

CLIENT NAME: XYZ COMPANY
MACHINE NAME: COMPRESSOR 2

TECHNICAL ASSOCIATES OF CHARLOTTE INC. (704) 333-9011

BY: JEB
DATE: 5/12/93

SPECTRAL ALARM BAND COMPUTER ENTRY FORM

POSITIONS	UNITS	ALARM	RPM	BRGS.	F_{MAX}	PARAMETER	BAND 1	BAND 2	BAND 3	BAND 4	BAND 5	BAND 6
1A, 1H, 1V 2A, 2H, 2V (From Case A)	mm/sec	5.50	1180	Ball	72K	LOWER FREQ. HIGHER FREQ. BAND ALARM	600 1440 5.00	1440 2640 2.25	2640 3840 1.50	3840 14,640 1.25	14,640 36,000 1.00	36,000 72,000 0.75
1 BAR (From Case E)	mm/sec	5.50	1180	Ball	360K	LOWER FREQ. HIGHER FREQ. BAND ALARM	30,000 85,000 1.00	85,000 140,000 1.00	140,000 195,000 0.85	195,000 250,000 0.80	250,000 305,000 0.70	305,000 360,000 0.60
2 EL (From Case F)	mm/sec	5.50	1180	Ball	12K	LOWER FREQ. HIGHER FREQ. BAND ALARM	240 2000 5.00	2000 4000 2.25	4000 6000 2.25	6000 8000 2.00	8,000 10,000 1.75	10,000 12,000 1.25
1H, 2H, 3H, 4H	gSE	0.50	1180	Ball	___	LOWER FREQ. HIGHER FREQ. BAND ALARM	NO ALARM BANDS FOR SPIKE ENERGY SPECTRA NOW					
3A, 3H, 3V 4A, 4H, 4V (From Case A)	mm/sec	6.50	1180	Taper Roller	72K	LOWER FREQ. HIGHER FREQ. BAND ALARM	600 1440 5.75	1440 2640 2.50	2640 3840 2.00	3,840 14,640 1.50	14,640 36,000 1.25	36,000 72,000 1.00
3HI - A&H 5HI - A&H (From Case C, Looking at GMF_1 (1st Mesh))	mm/sec	6.50	1180	Taper Roller	200K	LOWER FREQ. HIGHER FREQ. BAND ALARM	14,750 44,250 6.00	44,250 73,750 4.75	73,750 103,250 1.75	103,250 132,750 3.50	132,750 162,250 1.75	162,250 200,000 2.75
5A, 5H, 5V 6A, 6H, 6V (From Case A)	mm/sec	6.50	2950	Taper Roller	144K	LOWER FREQ. HIGHER FREQ. BAND ALARM	1440 3540 5.75	3540 6490 2.50	6490 9440 2.00	9440 36,000 1.50	36,000 72,000 1.25	72,000 144,000 1.00
*6HI - A&H 8HI - A&H (From Case C, Looking at GMF_2 (2nd Mesh))	g Accel.	10.0 g	2950	Taper Roller	540K	LOWER FREQ. HIGHER FREQ. BAND ALARM	39,800 119,500 2.0g	119,500 199,100 8.0g	199,100 278,800 3.0g	278,800 358,400 6.0g	358,400 438,100 3.0g	438,100 540,000 4.5g
5H, 6H	gSE	1.25	2950	Taper Roller	___	LOWER FREQ. HIGHER FREQ. BAND ALARM	NO ALARM BANDS FOR SPIKE ENERGY SPECTRA NOW					
7A, 7H, 7V 8A, 8H, 8V (From Case A)	mm/sec	6.50	5310	Taper Roller	300K	LOWER FREQ. HIGHER FREQ. BAND ALARM	3000 6400 5.75	6400 11,800 2.50	11,800 17,000 2.00	17,000 65,000 1.50	65,000 150,000 1.25	150,000 300,000 1.00
7H, 8H	gSE	1.90	5310	Taper Roller	___	LOWER FREQ. HIGHER FREQ. BAND ALARM	NO ALARM BANDS FOR SPIKE ENERGY SPECTRA NOW					
9A, 9H, 9V 10A,10H,10 (From Case G, Type 3 with 15 Vanes)	mm/sec	5.50	5310	Sleeve	108K	LOWER FREQ. HIGHER FREQ. BAND ALARM	1080 4250 1.10	4250 9560 5.00	9560 20,180 2.20	20,180 73,280 1.50	73,280 86,000 3.75	86,000 108,000 2.00

*Positions 6HI and 8HI must be Stud or Adhesive Mounted for these High Freq. Acceleration Measurements (to 540,000 CPM).

© Copyright 2001 Technical Associates Of Charlotte, P.C.

7.16 PERIODIC REEVALUATION OF SPECTRAL ALARM BAND SETUPS ON EACH FAMILY OF MACHINES

Table III gives the procedure on how to specify spectral alarm bands for a number of machine types and bearing configurations. It is intended for use when either originally setting up a PMP database before data is taken, or on the other hand, is intended as a starting point for those who have already taken large numbers of spectral measurements, but have never inserted alarm bands. For those who have specified bands, it serves as a reference to which these setups can be compared.

After baselines and perhaps after several additional surveys are completed with these bands in place, the user should reexamine each of the spectral alarm band setups on each family of machines. That is, he should identify certain families, or groupings of similar machines and evaluate how such bands are working, preferably using statistical analysis (24,32, 33,37). He should also evaluate how well the overall alarm levels he has specified for each machine point are functioning. He should ask such questions as:

1. Are the specified alarm bands detecting those problems which are occurring in this machine family - or are they catching some, but missing others?

2. If they are missing some problems, how should they be adjusted in order to better detect them?

3. On the other hand, are these alarm bands causing numerous “false alarms”? If this is allowed to continue, both those in the plant as well as analysts in the program will begin to pay little attention to so-called alarm conditions, even when they actually do detect potentially serious problems.

4. If alarm bands are causing false alarms, how can they be carefully adjusted to noticeably reduce the probability of sounding false alarms, but at the same time, not to the point where they will miss genuine problems?

5. For a given machine family, is one set of spectral alarm bands effectively serving the whole family, or is it necessary to further subdivide the family into two or more groupings? For example, if a utility specified one set of alarm bands for all fans, he will likely find that he needs to further subdivide fans into several individual groups including one set for ID fans, another for FD fans, another for primary air fans, and even another for gas recirculating fans (29).

Finally, it may even be necessary to have one set of alarm bands for the driver and another set for driven pieces of equipment (as was done in Table III, for example, on motor-driven pumps). Or, due to a particular vibration behavior for another family, he may have to specify one set of alarm bands for axial measurements and another for radial (and possibly even different alarm bands altogether for axial, horizontal and vertical measurements).

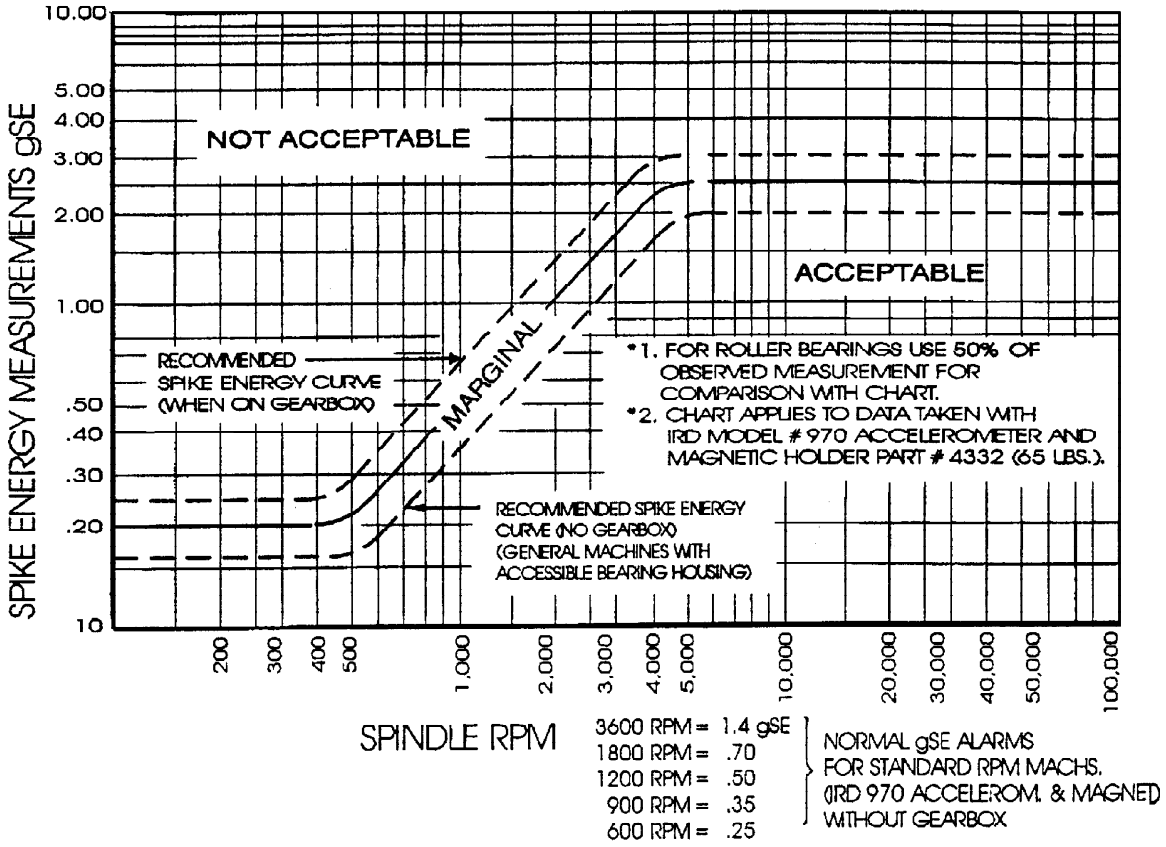
In general, closely examine each set of spectral alarm bands for each machine family on a quarterly basis (maximum 6 month intervals). Remember, these spectral alarm bands are your **“life-line”** with the specific purpose of detecting machine anomalies when they occur, and doing so early in the failure process so that corrective measurements can be orderly planned and so that required replacement parts can be acquired. Properly specified, these spectral alarm bands can and will save one many a headache (not to mention considerable dollar savings as well).

© Copyright 2001 Technical Associates Of Charlotte, P.C.

CHART 1
RECOMMENDED SPIKE ENERGY SEVERITY CHART (IRD SPIKE ENERGY)

FIGURE 7. RECOMMENDED SPIKE ENERGY SEVERITY CHART

SEVERITY CHART GUIDELINES FOR BALL BEARINGS*



SPIKE ENERGY™ MEASUREMENTS

Energy is generated by repetitive transient mechanical impacts. Such impacts typically occur as a result of surface flaws in rolling-element bearings or gear teeth.

This energy is conducted from its source through various paths to the outer surface of the machine structure, and is seen as a small-amplitude vibration at the surface. Accelerometers coupled to the surface generate corresponding electrical signal.

The accelerometer signals processed by unique filtering and detection circuitry to produce a single "figure of merit" related to the intensity of the original impacts. This figure of merit is expressed in "gSE" units.

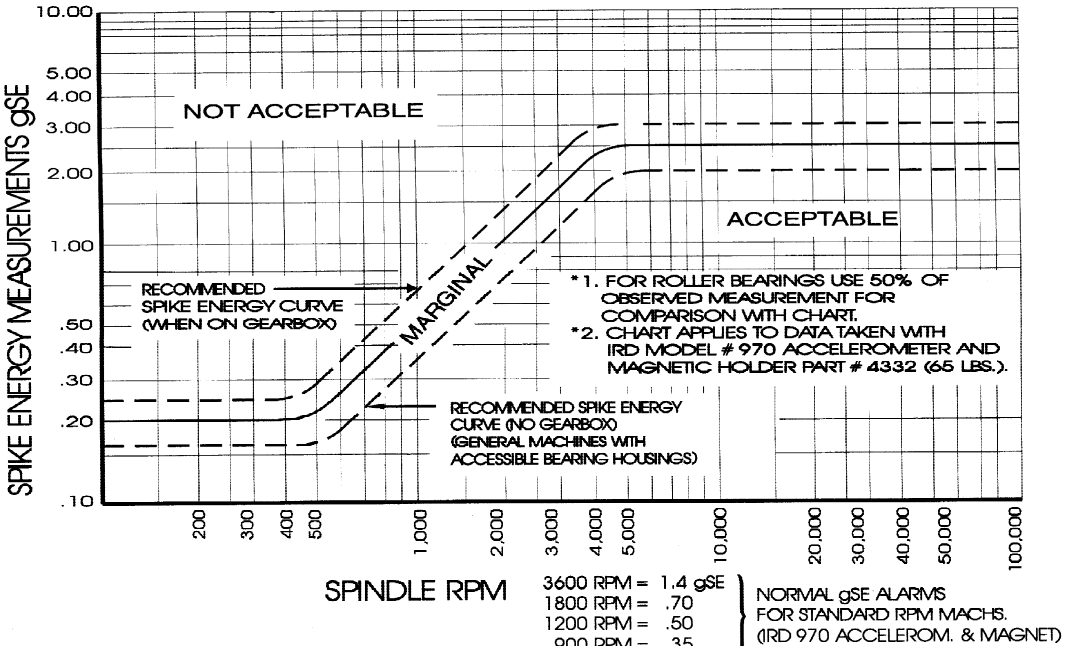
SPIKE ENERGY™ gSE readings are measurements which can with experience, be correlated with the severity of the casual surface flaws. Even though gSE readings are affected by the nature of the conductive path between the impact source and the accelerometer, similar machine structures will provide a reasonable basis for comparison between the structures.

The gSE figure of merit has proven to be effective in detecting mechanical defects in meshing gears and rolling element bearings. The gSE measurement, when used in conjunction with conventional measurement of vibration velocity and acceleration, provides early indications of mechanical deterioration.

** When used with magnetic holders, accelerometers must be installed with a light coating of silicone grease and tightened to 40 in-lb. torque.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

CHART 2
MAINTENANCE DIAGNOSTICS VIBRATION AND HIGH FREQUENCY GENERAL TOLERANCE CHART FOR PROCESS MACHINERY WITH ROLLING ELEMENT BEARINGS①



① Provided by Charles Berggren of Monsanto
② Spike Energy Amplitudes measured using an IRD 970 accelerometer outfitted with IRD's 2-pole, 65 lb. magnet.

7.161 Procedure For Evaluating the Effectiveness of Specified Overall Alarm Levels and Spectral Bands:

Statistical analysis should be employed on both overall alarm levels and spectral band setups for each family of machines. The advent of the personal computer and statistical software now allows one to perform powerful statistical analysis on large numbers of data points that formerly would have been impractical (24,32,33).

The procedure for evaluating overall alarm levels is summarized as follows:

1. Take each family of machines and work with the latest data captured on each machine in that family (for example, a utility might work with the data taken on each point of the ID fan family).

2. Calculate the average overall level for the family (X_{ave}) by summing the level for each point on each machine in the entire family and dividing by the total number of samples as per the following formula:

$$X_{ave} = \frac{\sum_{i=1}^{n} X_i}{n} = \frac{X_1 + X_2 + X_3 + \ldots + X_n}{n} \qquad \textbf{EQUATION (2)}$$

Where:

X_{ave} = Average or Mean Value
X_i = Value of each individual sample i
n = Number of Samples

© Copyright 2001 Technical Associates Of Charlotte, P.C.

3. Calculate the Standard Deviation (S) for the data in this family as per Equation (3):

$$S = \sqrt{\frac{\sum_{i=1}^{n}(X_i - X_{ave})^2}{n-1}} = \sqrt{\frac{(X_1 - X_{ave})^2 + (X_2 - X_{ave})^2 + \ldots + (X_n - X_{ave})^2}{n-1}} \qquad \textbf{EQUATION (3)}$$

Where:

S = Standard Deviation

4. Then, calculate the Statistical Overall Alarm for the family using the following formula:

Statistical OA = X_{ave} + 2S **EQUATION (4A)**

(some prefer OA = X_{ave} + 3S) **EQUATION (4B)**

Assuming a normal distribution, approximately 95.5% of the data should fall within 2 standard deviations of the statistical average, whereas 99.7% should fall within 3 standard deviations. As the notes accompanying Equations 4A and 4B state, research and application has shown that a statistical overall equal to only the average level (X_{ave}) plus 2 Standard Deviations should be used during Baselining and the first few follow-up surveys before the plant has had a chance to take corrective actions (since vibration levels will be higher at that time. Later, after corrective actions have been taken (which should lower vibration levels), Equation 4B should be used adding 3 Standard Deviations to the mean value for the Statistical Overall.

5. Compare the formerly specified overall for this family of machines to the calculated statistical overall alarm. If significantly different, normally choose an alarm level close to the statistical alarm since it is based on actual measurements on your specific family of machines. However, if it is felt that the current statistical alarm was calculated from an extraordinarily high number of machines either in very good condition, or vice versa, in very bad operating condition, then go with an overall alarm between the specified and statistical, but weighted towards the statistical. (Please see the example case history in Table IV which includes several “aberrations” out of the 25 different machine “families” investigated at 4 different plants).

This same procedure can be used for evaluating the effectiveness of each of the spectral alarm bands, whether using 6 bands, or up to 400 in those systems which allow narrowband alarms. In these cases, take one band at a time; calculate the average for the entire family; then calculate the standard deviation for that band; then calculate the statistical level for that band adding either two or three standard deviations to the mean value as desired. Then, compare the calculated statistical level to the alarm level for that band which had formerly been specified, and adjust as desired. Computer software is available for automating this statistical analysis process.

7.162 EXAMPLE - “Statistical Analysis of Overall Vibration Velocity in 4 Client Power Plants Using the Procedure Recommended Above”:

Table IV includes results of a statistical analysis performed on overall velocity levels from 25 different machine types (“families”) in each of 4 power plants for a client. Data from 442 machines were analyzed as per the statistical procedure outlined above. Not only were the overall levels themselves analyzed, but also the spectral band setups for each of the 25 machine families as well. However, for brevity, only the analysis of overalls is presented here.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

Referring to Table IV, note that each of 4 items were tabulated for each plant. First, the quantity of machines in each family was recorded. Next, the overall alarm level used by this plant on each machine family was documented. Then, the average overall for each family was calculated. As few as 5 data collection points were on some machines including motor-mounted fans, whereas as many as 40 data collection points were on some turbine/generators. At this point, no effort was made to separately analyze axial and radial readings, but this step is now being taken. Note that the analysis did separately evaluate turbine/generator gearboxes and exciters from the turbine/ generators themselves. Finally, note that the standard deviation was computed for each of the average overall levels.

In general, the study showed that overall alarms used by the client during his baselining were within reason. However, adjustments had to be made to a few.

First, 2 families were found to have excessively high vibration which resulted in high statistical alarms. The 8 turbine/generator gearboxes analyzed had an average .423 in/sec overall and a very high standard deviation of .248 in/sec which resulted in a statistical alarm of .919 in/sec. Similarly, the 7 motor-mounted fans had an average .310 in/sec overall and a high .196 in/sec standard deviation (giving a .701 in/sec statistical alarm using 2 standard deviations). The actual spectrum analysis performed on each of these 2 families of machines found very serious problems in a large proportion of them. Therefore, it was felt that the measured amplitudes were unrealistically high. In the case of the 8 turbine/generator gearboxes, the “recommended alarm” (in the far right-hand column of Table IV) was set at .500 in/sec (up from .440 in/sec). Similarly, the motor-mounted alarm was increased slightly from .325 to .375 in/sec.

On the other hand, several families had much lower vibration than the overall alarms specified by the client. This included each of 7 turbine/generators operating at 1800 RPM; 12 turbine/ generators operating at 3600 RPM; 45 vertical condensate/circulating water pumps; 32 forced draft fans; 32 induced draft fans; and 9 general purpose gearboxes. For example, in the case of the turbine/generators, the client used the overall alarms now being proposed by the latest version of ISO 2372 now under consideration which it is understood will call for .300 in/sec peak for 1800 RPM machines and .440 in/sec peak for 3600 RPM turbine/generators.

Here, the average levels were only .097 in/sec for 1800 RPM machines and .147 in/sec for the 3600 RPM turbine/generators. Each of these families had fairly low average standard deviations of overalls which resulted in statistical alarms of .226 in/sec for the 1800 RPM and .339 in/sec for the 3600 RPM turbine/ generators (see Table IV). Therefore, recommended alarm levels for both machines were dropped from .300 in/sec to .250 in/sec for the 1800 RPM machines and from .440 in/sec to .350 in/sec on the 3600 RPM machines. These will be further adjusted, if needed, in the future. Similar steps were taken on the vertical condensate/circulating water pumps, forced draft fans, induced draft fans and general purpose gearboxes.

One family obviously had statistically poorly grouped readings with an average overall for 18 general purpose fans being .175 in/sec compared to its standard deviation of .203 in/sec, far above the anticipated norm for a standard deviation compared to this average. Notice that standard deviation levels at both plant 3 and plant 4 were high on these machines. As a result, it was recommended to leave the alarm setting at .375 in/sec until further data has been recorded, and until several machines have been repaired (the spectrum analysis also found a number of problems in these machines which required corrective action).

Finally, please note the bottom row of Table IV entitled “TOTALS”. Notice that the average overalls for all machines at each plant were all fairly close to one another ranging from a low of .097 in/sec at Plant 2 up to a high of .128 in/sec at Plant 4. Plant 4 had the largest standard deviation of .120 in/sec. Interestingly, after analyzing data from thousands of points on these 442 machines, the final calculated statistical alarm for all of these machines came out to be a level of

© Copyright 2001 Technical Associates Of Charlotte, P.C.

.294 in/sec - very close to the so-called “industry standard” of .300 in/sec which many plants employ as an alarm on all families of machines, regardless of their machine type, speed or drive configuration. However, by examining Table IV, there still was a great disparity between the average overalls of some families.

7.17 CONCLUSIONS

The last few years have brought about dramatic accomplishments and innovative new, powerful tools into the fields of vibration analysis and predictive maintenance. In addition, the rapid development of the personal computer continues to provide the creative team of vibration analyst and computer programmer with an exciting new menu of imaginative ways of not only storing and processing the vast amount of data being gathered, but of effectively putting it to use. This has the effect of multiplying the accomplishments of those within the vibration diagnostics field, and goes far in expanding the horizons of our knowledge. Surely, many more machine “problem sources” will be added to the diagnostic chart of Table I within just the next 5 years, and some of the information now in the chart will be refined. When this occurs, one will be enabled to set up even more effective spectral alarm bands than those provided in Table III. No matter what happens, the effectiveness of every program will depend on the capability of that system to first detect the presence of real problems, and then all available means including software and the knowledge of vibration analysts themselves will be deployed to diagnose both the cause and severity of the problems. Therefore, it now appears that the success of all future systems will depend on how effective the spectral alarm band setups will be in accomplishing this critical problem detection process. It is sincerely hoped that this paper will be of some assistance in helping many to compose exactly this quality of effective spectral alarm bands.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

TABLE IV
STATISTICAL ANALYSIS OF OVERALL VIBRATION VELOCITY IN 4 CLIENT POWER PLANTS @ 3/28/90

MACHINE TYPE	PLANT 1				PLANT 2				PLANT 3				PLANT 4				CORPORATE SUMMARY (4 PLANTS)					
	QTY.	OA ALARM	OA AVE.	STD. DEV.	QTY.	OA ALARM	OA AVE.	STD. DEV.	QTY.	OA ALARM	OA AVE.	STD. DEV.	QTY.	OA ALARM	OA AVE.	STD. DEV.	QTY.	OA ALARM	CORP. AVE. OA	CORP. AVE. S.D.	STAT. ALARM	RECOM. ALARM
TURBINE GENERATORS																						
A) 1800 RPM	--	--	--	--	2	.300	.078	.041	5	.300	.105	.074	--	--	--	--	7	.300	.097	.065	.226	.250
B) 3600 RPM	4	.440	.161	.093	4	.440	.139	.092	3	.440	.158	.119	1	.440	.084	.055	12	.440	.147	.096	.339	.350
TURB/GEN. GEARBOXES	4	.440	.286	.291	2	.440	.401	.157	2	.440	.718	.253	--	--	--	--	8	.440	.423	.248	.919	.500
TURB/GEN. EXCITERS	4	.300	.153	.082	6	.250	.044	.049	5	.250	.092	.111	--	--	--	--	15	.263	.089	.078	.246	.250
BOILER FEED PUMPS																						
A) MOTOR	4	.300	.113	.067	6	.300	.141	.078	9	.375	.155	.092	--	--	--	--	24	.328	.136	.079	.294	.300
B) TURBINE	--	--	--	--	2	.300	.148	.079	--	--	--	--	2	.300	.233	.127	4	.300	.191	.103	.397	.375
BOOSTER PUMPS	--	--	--	--	2	.300	.087	.032	2	.300	.056	.043	2	.300	.144	.049	6	.300	.096	.041	.178	.200
COND/CIRC WATER PUMPS																						
A) HORIZONTAL	--	--	--	--	--	--	--	--	--	--	--	--	2	.300	.090	.054	2	.300	.090	.054	.198	.200
B) VERTICAL	8	.300	.040	.015	11	.300	.065	.092	18	.300	.081	.071	8	.300	.069	.047	45	.300	.068	.062	.191	.200
HOTWELL PUMPS	7	.350	.140	.083	7	.300	.074	.125	7	.300	.064	.051	--	--	--	--	21	.317	.093	.086	.265	.300
SLAG SLUICE PUMPS	--	--	--	--	--	--	--	--	6	.300	.134	.088	--	--	--	--	6	.300	.134	.088	.310	.300
GENERAL PURPOSE PUMPS	12	.300	.128	.092	10	.300	.081	.054	41	.300	.094	.069	21	.300	.127	.123	84	.300	.106	.084	.274	.300
CENTRIF. AIR COMPRESSORS	1	.300	.125	.053	--	--	--	--	--	--	--	--	2	.250	.102	.070	3	.267	.110	.064	.238	.250
RECIP. AIR COMPRESSORS	5	.425	.149	.054	7	.425	.047	.023	10	.425	.197	.214	4	.425	.125	.075	26	.425	.136	.110	.357	.425
FORCED DRAFT FANS	8	.375	.096	.078	8	.375	.060	.040	12	.375	.063	.054	4	.375	.066	.048	32	.375	.071	.056	.182	.250
INDUCED DRAFT FANS	8	.375	.120	.068	8	.375	.045	.024	12	.375	.066	.046	4	.375	.063	.060	32	.375	.074	.048	.169	.200
MOTOR-MOUNTED FANS	--	--	--	--	--	--	--	--	5	.275	.255	.165	2	.375	.446	.272	7	.325	.310	.196	.701	.375
PRIMARY AIR FANS	--	--	--	--	--	--	--	--	11	.375	.132	.159	--	--	--	--	11	.375	.132	.159	.450	.375
ROOTS BLOWERS	--	--	--	--	--	--	--	--	4	.375	.110	.066	--	--	--	--	4	.375	.110	.066	.242	.300
FLY ASH BLOWERS	--	--	--	--	2	.375	.236	.107	--	--	--	--	--	--	--	--	2	.375	.236	.107	.450	.375
AIR HEATERS	8	.375	.070	.056	--	--	--	--	8	.375	.080	.059	--	--	--	--	16	.375	.075	.058	.190	.250
GENERAL PURPOSE FANS	--	--	--	--	--	--	--	--	6	.375	.212	.128	12	.375	.156	.241	18	.375	.175	.203	.581	.375
PULVERIZERS	--	--	--	--	14	.375	.147	.144	26	.375	.128	.094	--	--	--	--	40	.375	.135	.112	.358	.375
GEN. PURPOSE MACHINES	--	--	--	--	--	--	--	--	8	.300	.066	.032	--	--	--	--	8	.300	.066	.032	.130	.200
GEN. PURPOSE GEARBOXES	--	--	--	--	--	--	--	--	7	.400	.099	.080	2	.300	.126	.066	9	.378	.105	.077	.259	.250
TOTALS	**78**	**N/A**	**.121**	**.080**	**91**	**N/A**	**.097**	**.078**	**207**	**N/A**	**.116**	**.098**	**66**	**N/A**	**.128**	**.120**	**442**	**N/A**	**.115**	**.090**	**.294**	**N/A**

REFERENCES

1. American Gear Manufacturers Association (AGMA); Arlington, Virginia; AGMA Standard 110.04 (August, 1980); "Nomenclature of Gear Tooth Failure Modes"; Pages 6 - 23.

2. Bently Nevada Corporation; Minden, Nevada; Mechanical Engineering Seminar; October, 1984; "Section 6 - Introduction to Rotor Dynamics"; "Section 9 - Rotor Instability"; "Section 11 - Machinery Rubs".

3. Berggren, J. Charles; Monsanto Chemical Company; Pensacola, Florida; "Diagnosing Faults in Rolling Element Bearings, Part I. Assessing Bearing Condition"; Vibrations, Volume 4, No. 1; March, 1988; Pages 5 - 14.

4. Berggren, J. Charles; Monsanto Chemical Company; Pensacola, Florida; "Diagnosing Faults in Rolling Element Bearings, Part II. Alternative Analytical Methods"; Vibrations, Volume 4, No. 2; June, 1988; Pages 12 - 23.

5. Berggren, J. Charles; Monsanto Chemical Company; Pensacola, Florida; "Diagnosing Faults in Rolling Element Bearings, Part III. Electronic Data Collector Applications"; Vibrations, Volume 5, No. 2; June, 1989; Pages 8 - 19.

6. Berry, James E.; Technical Associates of Charlotte, Inc.; Charlotte, North Carolina; "Problem Diagnostics on High-Speed Centrifugal Compressors Using Vibration Signature Analysis"; Proceedings 12th Annual Meeting - The Vibration Institute; Nashville, Tennessee; May, 1988; Pages 1 - 13.

7. Bradbury, E. R.; Union Carbide Industrial Gases, Inc.; Tonawanda, New York; "The Control Chart: A Basic Tool of Statistical Quality Control"; Proceedings 13th Annual Meeting - The Vibration Institute; June, 1989; Pages 87 - 92.

8. Bradley, Dan; IRD Mechanalysis; Columbus, Ohio; "Introduction to FFT Terms and Parameters"; Pages 1 - 9.

9. Bruel & Kjaer; Marlborough, Massachusetts; The Application of Vibration Measurement and Analysis in Machine Maintenance; "The Application of Frequency Analysis to Machine Diagnosis"; Chapter 7, Pages 1 - 12.

10. Bruel & Kjaer; Marlborough, Massachusetts; Piezoelectric Accelerometers and Vibration Preamplifiers Theory and Application Handbook; March, 1978; Pages 50 - 59.

11. Campbell, W. R.; ARAMCO; Dhahran, Saudi Arabia; "Diagnosing Alternating Current Electric Motor Problems"; Vibrations, Volume 1, No. 3; December, 1985; Pages 12 - 15.

12. Corey, Cletus A.; Magnetek, Louis Allis; Milwaukee, Wisconsin; "Induction Motor Electrical Noise and Vibration - Sources and Case Problems"; Proceedings 12th Annual Meeting - The Vibration Institute; May, 1988; Pages 171 - 178.

13. Eshleman, Ronald L.; Vibration Institute; Clarendon Hills, Illinois; "Periodic Monitoring for Predictive Maintenance"; Vibrations, Volume 3, No. 1; June, 1987; Pages 3 - 8.

14. Eshleman, Ronald L.; Vibration Institute; Clarendon Hills, Illinois; "Techniques for the Development of Criteria and Limits for Monitoring Machinery Vibration"; Vibrations, Volume 2, No. 2; September, 1986; Pages 5 -11.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

15. Eshleman, Ronald L.; Vibration Institute; Clarendon Hills, Illinois; “Machinery Condition Analysis”; Vibrations, Volume 4, No. 2; June, 1988; Pages 3 - 11.

16. Hewlett-Packard Company; Palo Alto, California; Dynamic Signal Analyzer Application - Effective Machinery Maintenance Using Vibration Analysis, “Application Note 243-1”; October, 1983; Pages 23 - 44.

17. Hewlett-Packard Company; Palo Alto, California; The Fundamentals of Signal Analysis, “Application Note 243”; July, 1982; Pages 13 - 17, 25 - 39.

18. Hydraulic Institute; Cleveland, Ohio; “Acceptable Field Vibration Limits for Vertical Non-Clog Pumps”; Hydraulic Institute Standards for Centrifugal, Rotary & Reciprocating Pumps, 14th Edition, 1983; Figure 78, Page 121.

19. International Standard Organization; ISO 2372 - “Mechanical Vibration of Machines With Operating Speeds From 10 to 200 rev/sec - Basis For Specifying Evaluation Standards”; First Edition, 1974-11-01; Pages 1 - 7.

20. IRD Mechanalysis; Columbus, Ohio; Advanced Training Manual, “Vibration Analysis”; Pages 51 - 142.

21. IRD Mechanalysis; Columbus, Ohio; Vibration Technology - II; 1989; “Systems Dynamics & Resonance”.

22. IRD Mechanalysis; Columbus, Ohio; Vibration Technology - I; 1988; Pages 5-1 thru 6-20.

23. Jacobs, Ronald W.; Monsanto Company; Addyston, Ohio; “Detection of Mechanical Faults in Rotary Blowers”; Vibrations, Volume 2, No. 1; June, 1986; Pages 9 -13.

24. Jacobs, Ronald W.; Monsanto Chemical Company; Addyston, Ohio; “SQC And Predictive Maintenance”; Proceedings 13th Annual Meeting - The Vibration Institute; June, 1989; Pages 83 - 86.

25. Maxwell, J. Howard; Arizona Public Service Company; Palo Verde Nuclear Generation Station; Phoenix, Arizona; “Induction Motor Magnetic Vibration”; Proceedings Machinery Vibration Monitoring and Analysis Meeting - The Vibration Institute, The Vibration Institute, April, 1983, Pages 39 - 51.

26. Middleton; Ben; Palomar Technology International; Carlsbad, California; “Rolling Element Bearing Failure Detection Methods”; Presented at the Acoustical Society of America, Raleigh, North Carolina, October 8 - 9, 1987; Pages 1 - 14.

27. Mitchell, John S.; Palomar Technology International; Carlsbad, California; An Introduction To Machinery Analysis and Monitoring; Pennwell Publishing Company; Tulsa, Oklahoma; 1981; Pages 141 - 151, 172 - 204.

28. Peterson, David; Computational Systems, Inc.; Knoxville, Tennessee; “Vibration Alarm Methods in Predictive Maintenance Programs”; P/PM Technology Volume 3, Issue 1 - January/February, 1990; Pages 22 - 25.

29. Piety, Kenneth R.; Piety, Richard W.; Computational Systems, Inc.; Knoxville, Tennessee; Scheibel, John R. (Electric Power Research Institute); “Vibration Monitoring of Centrifugal Fans in Fossil-Fired Power Generation”; Vibrations, Volume 6, No. 1; March, 1990; Pages 8 - 13.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

30. Piotrowski, John D.; General Electric Company; Evendale, Ohio; “Alignment Condition and Its Effect on The Vibration Response of Rotating Machinery”; Vibrations, Volume 1, No. 4; March, 1986; Pages 11 - 17.

31. Rockland Scientific Corp.; Rockleigh, New Jersey; “Machinery Vibration Diagnostic Guide, Application Note 22”.

32. Smiley, R. G.; Entek Scientific Corporation; Cincinnati, Ohio; “Set Alarm Levels Without Guesswork”; Proceedings 12th Annual Meeting - The Vibration Institute; May, 1988; Pages 107 - 112.

33. Smiley, R. G. and Schlitz, R. L.; Entek Scientific Corporation; Cincinnati, Ohio; “Statistics or Standards?”; Sound and Vibration; September, 1989; Pages 22 - 23.

34. Szrom, David B.; Mechanical Consultants, Inc.; Homewood, Illinois; “Analysis and Correction of Gearbox Defects”, Proceedings Machinery Vibration Monitoring and Analysis Meeting - The Vibration Institute, June, 1984; Pages 147 - 153.

35. Szrom, David B.; Mechanical Consultants, Inc.; Homewood, Illinois; “Determining Gear Condition With FFT Spectrum Analysis”; Proceedings 11th Annual Meeting - The Vibration Institute; June, 1987; Pages 1 - 5.

36. Thomson, W. T. and Chalmers, S. J.; Robert Gordon’s Institute of Technology; Aberdeen, Scotland; “An On-Line Computer Based Current Monitoring System for Rotor Fault Diagnosis and 3-Phase Induction Motors”; Pages 1 - 15.

37. Wetzel, Rick; Entek Scientific Corporation; Cincinnati, Ohio; “Statistical Alarm Methods”; Pages 1 - 4.

38. Winterton, John G.; Bently Nevada Corporation; Broomall, Pennsylvania; “Component Identification of Gear Generated Spectra”; Proceedings 11th Annual Meeting - The Vibration Institute; June, 1987; Pages 11 - 17.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

CHAPTER 8

COMMON PITFALLS IN EVERYDAY VIBRATION MEASUREMENT

8.0 INTRODUCTION

Good quality data is essential to every vibration program. Without proper representation of the actual vibration, the diagnostic results, conclusions and recommendations become meaningless. The following pointers are given to help avoid pitfalls which can plague a vibration PM program, sometimes rendering it almost ineffective.

8.1 GENERAL CONSIDERATION FOR OBTAINING CONSISTENT QUALITY DATA

The following observations are provided based on many years of vibration data collection field experience. Also, they are broken into categories which reflect the order in which PM programs or vibration data collection would be conducted.

8.11 CHOOSING MEASUREMENT LOCATIONS

1) Choose proper measurement locations. Figure 1 shows both good and poor measurement locations. Locations should be as close to the machine bearings as possible. Try to locate these points within the load zone of the bearing, particularly on sleeve bearing supported machines, as well as heavily loaded or low-speed rolling element bearing machines. Studies have shown that placing the pickup within the load zone of low-speed, heavy rolls can increase the measured amplitude by 100% or more.

2) Do not compromise safety for a vibration measurement. Install a permanently mounted transducer if necessary to prevent potential injury.

3) Take horizontal measurements as near to the horizontal shaft centerline as possible (parallel to the earth). Also, see that the transducer is mounted as close to the bearing as possible, particularly if high frequency banded acceleration measurements (such as spike energy) are to be taken. Here again, please see Figure 1.

4) Take vertical measurements in as close to the vertical shaft centerline as possible (plumb with respect to the earth). Also, see that the transducer is mounted as close to the bearing as possible.

5) Take axial measurements parallel to the shaft being supported by the bearing and make the measurement at the same location each time (see Figure 1). Do not take axial data at the 12:00 position on one survey; then at 3:00; 6:00; or 9:00 on successive surveys. Again position the transducer as close to the bearing as possible.

6) On pumps, do not mistake seal locations for bearing locations.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

7) Do not take bearing measurements on a foundation or fabricated bases unless that is truly what you want.

8) Do not assign measurement locations to thin/weak sheet metal such as on some motor end bells. It is preferable in this case to take measurements on a substantial part of the motor such as the motor case (see Figure 1).

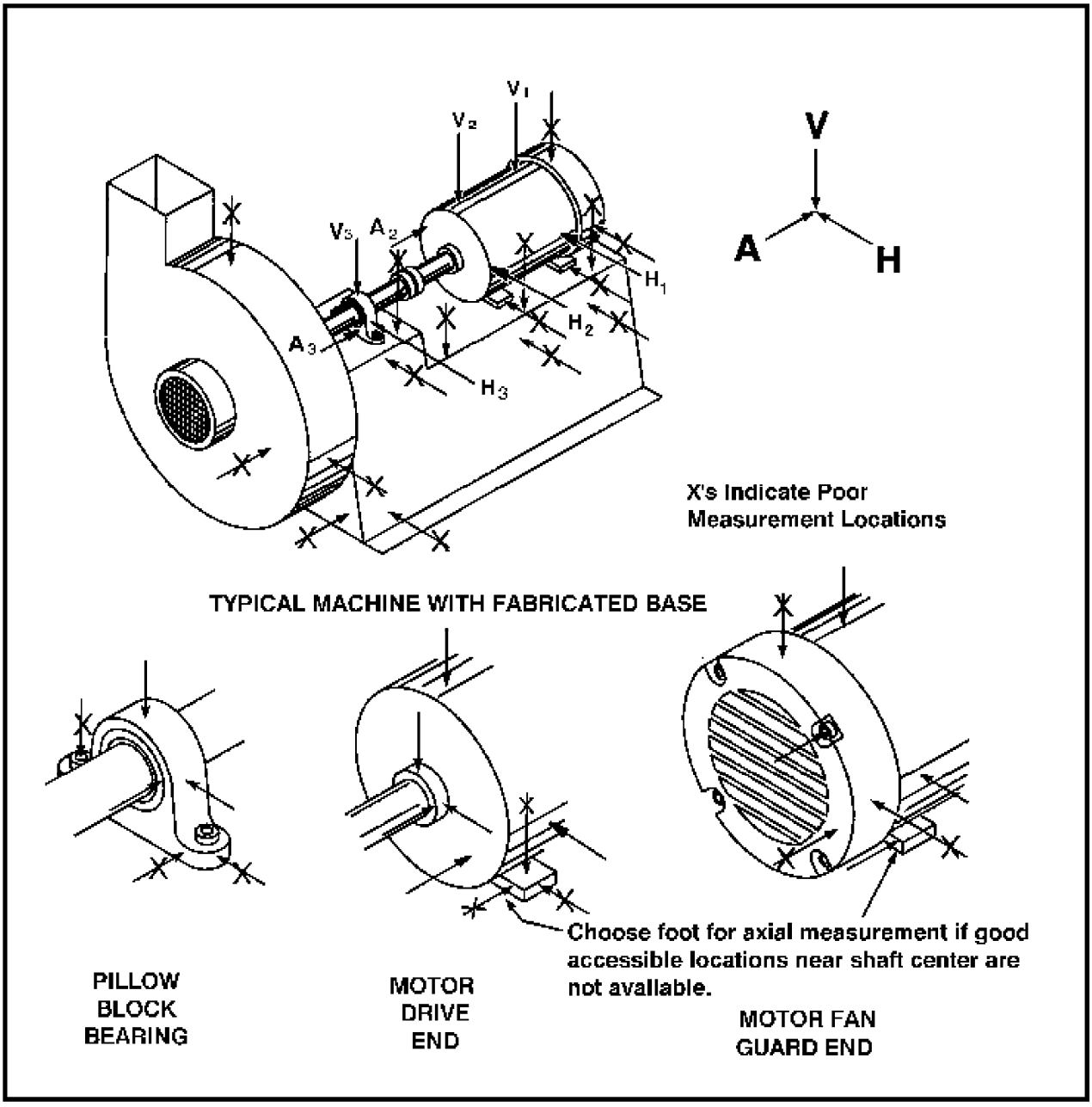


DIAGRAM SHOWING GOOD AND POOR MEASUREMENT LOCATIONS

FIGURE 1

© Copyright 2001 Technical Associates Of Charlotte, P.C.

8.12 MACHINE AND POINT IDENTIFICATION

1) Adopt naming and position identification conventions for all machines and points to be included in the PM program. Remain consistent as one moves from one plant area to another with these naming conventions. Table 1 shows one such naming convention. Please keep the following items in mind when adopting naming conventions and physically marking them on the machinery:

A) Number the measurement locations beginning from the physical outboard motor bearing to the outboard driven machine bearing (as shown in Figure 2). For example, assume a motor-driven fan. It is recommended that one of the following conventions be adopted as shown in Table 1 (or something similar):

TABLE I.
RECOMMENDED OPTIONAL MEASUREMENT LOCATION NAMING CONVENTIONS

LOCATION ON THE MACHINERY	NAMING CONVENTIONS	
	OPTION 1	OPTION 2*
MOTOR OUTBOARD BEARING (FAN SIDE)	1	MOV
MOTOR INBOARD BEARING (DRIVE SIDE)	2	MIV
FAN (DRIVEN) INBOARD BEARING	3	FIV
FAN (DRIVEN) OUTBOARD BEARING	4	FOV

*NOTE: OPTION 2 ASSUMES VELOCITY MEASUREMENTS. IF ACCELERATION IS MEASURED INSTEAD, AN "A" WOULD BE ENTERED AS THE THIRD LETTER.

Insert additional points in between these for equipment such as multi-stage gearboxes as needed.

B) Use consistent lettering conventions for the direction of the pickup (transducer) such as the following:

NAME	MEASURING DIRECTION
A	Axial
H	Horizontal
V	Vertical

C) If measuring on a vertically oriented machine, such as a vertical pump, both of the radial measurements at each bearing will be in the horizontal direction. In this case it is necessary to know the north/south and east/west directions at the machine. If your PMP software will allow, input the point names as 1N, 1E (for bearing 1 in the north and east directions). Point 1A will be parallel to the shaft (or perpendicular to the earth). However, some PMP software only allows a point to be named with A, H, and V. Therefore, adopt a naming convention such as H being the east direction and V being the north direction. Usually, there is another input field in the PMP software for entering a point description. Use this field to tell the user which direction to measure. For instance, for point 1H, input “1 East” in the point description field and set the data collector to display this field when collecting data on a route.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

2) Clearly mark the machine names with the name seen in the PMP computer database and data collector itself. This may require the use of a special paint which will withstand the environment in which the machine is located. On the other hand, if the machines themselves are periodically painted, or if the environment is too harsh, permanent metal stamps with the machine name imbedded on them may have to be used.

3) Clearly mark and label measurement point locations on each machine. Here again, metal stamps with the point identification may have to be used. Also, a washer may be permanently mounted (welded or glued) to the machine. This will not only provide a clearly located measurement point for consistent vibration data collection but also provide a good flat surface to mount the transducer or magnetic mount without worrying about it rocking on a rounded machine surface during data collection thereby rendering the data useless.

4) Provide machine drawings for each PMP machine showing measurement locations. See the sample machine drawing in Figure 2. Include important machine configuration information on the sketch such as bearing type and model number, gear tooth count, operating speed ranges of each component (RPM), blade or vane count on pumps/ blowers, etc.

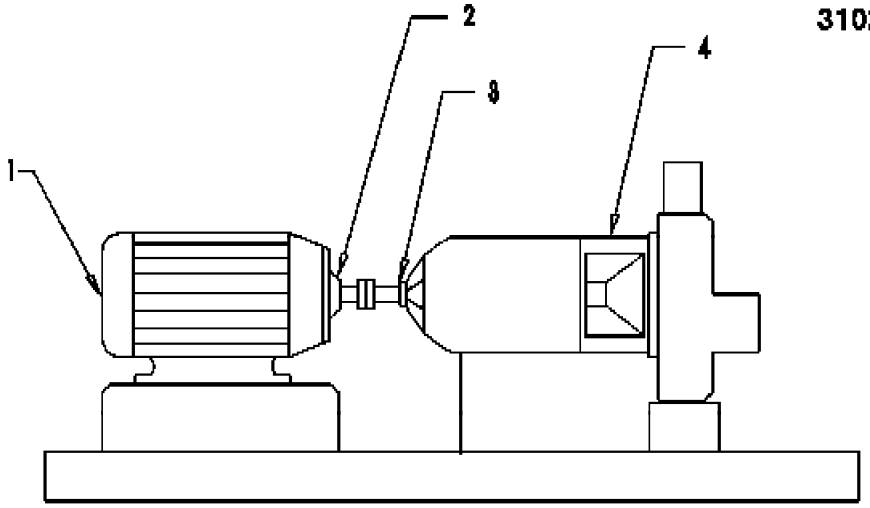
5) Construct a map of PMP machine locations within the plant. Figure 3 is a sample machine location map. This will help the person collecting data to verify that he is at the correct machine when collecting data as well as helping him set up an efficient route. This is crucial when many similar pieces of machinery are present in different locations in the plant such as air handlers or pumps. Typically, they are given the same name differentiated only by an equipment number. Make sure that the number is on the machine, in the PMP database, and on the machine location map.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

Technical Associates of Charlotte
(704)333-9011

MACHINERY DATA GUIDE SHEET

RELIABILITY SURVEY predictive maintenance program (PAGE___OF___)

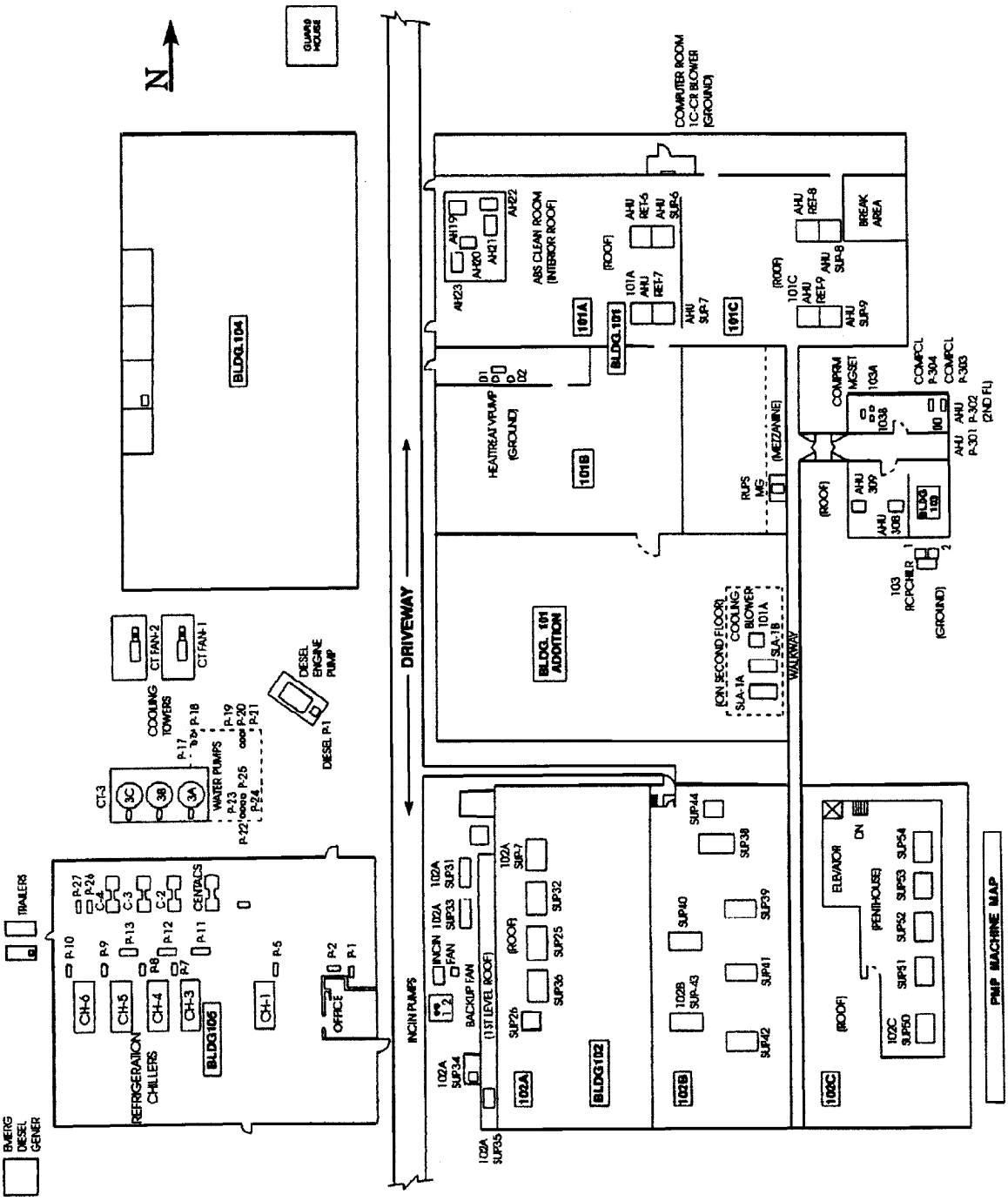


MACHINE IDENT.: 3 COND XFER PUMP **AREA:** NGS-3
ROUTE:

DRIVER	INTERMEDIATE	DRIVEN
RPM min: 3560	RPM in:	RPM min: 3560
max:	out:	max:
MFGR: WESTINGHOUSE	RATIO:	MFGR: WESTINGHOUSE/PACIFIC
MODEL: 7209	MODEL:	MODEL:
SER NO.:	SER NO.:	SER NO.:
V 480 P 3 A 109	RATING:	RATING:
SHEAVE: DIA.	BELT# :	# BLADES:
BRG# : BALL 6312 (I)	BRG# : (I)	BRG# : BALL (I)
BRG# : BALL 6312 (O)	BRG# : (O)	BRG# : BALL (O)
H.P.: 100	GEAR T: CT.	GEAR T: CT.
OTHER:	OTHER:	OTHER:

©Copyright 1992 Technical Associates of Charlotte, Inc.

FIGURE 2
SAMPLE MACHINE DRAWING



SAMPLE MACHINE LOCATION MAP
FIGURE 3

© Copyright 2001 Technical Associates Of Charlotte, P.C.

8.13 MEASUREMENT PARAMETERS

1) Specify the optimum frequency span for each measurement point on each PMP machine. Guidelines recommending such frequency ranges will be given in Chapter 7. It is very important to know the vibration source frequencies such as bearing, gear mesh, blade pass, electrical line, and rotor bar pass frequencies that may occur with each machine. Also, harmonics of these frequencies must be considered. In general, if there is no gearbox involved, rolling element bearing machines would require F_{MAX} at about 40X to 50X RPM, while sleeve bearing machines would require a maximum frequency of only about 20X RPM (assuming speeds are above about 1500 RPM).

However, there will be special cases requiring much higher frequency spans. For example, consider the gear-driven blower shown in Figure 4. This blower is driven by a motor running at 3580 RPM. However, a gear unit is used to increase the blower shaft speed to 6086 RPM. Because one gear has 68 teeth and the other has 40 teeth, the gear mesh frequency (# teeth x RPM) is 243,440 CPM. Also, an SKF 22212 bearing is used on the output gear shaft which has bearing fault frequencies as noted in Figure 4.

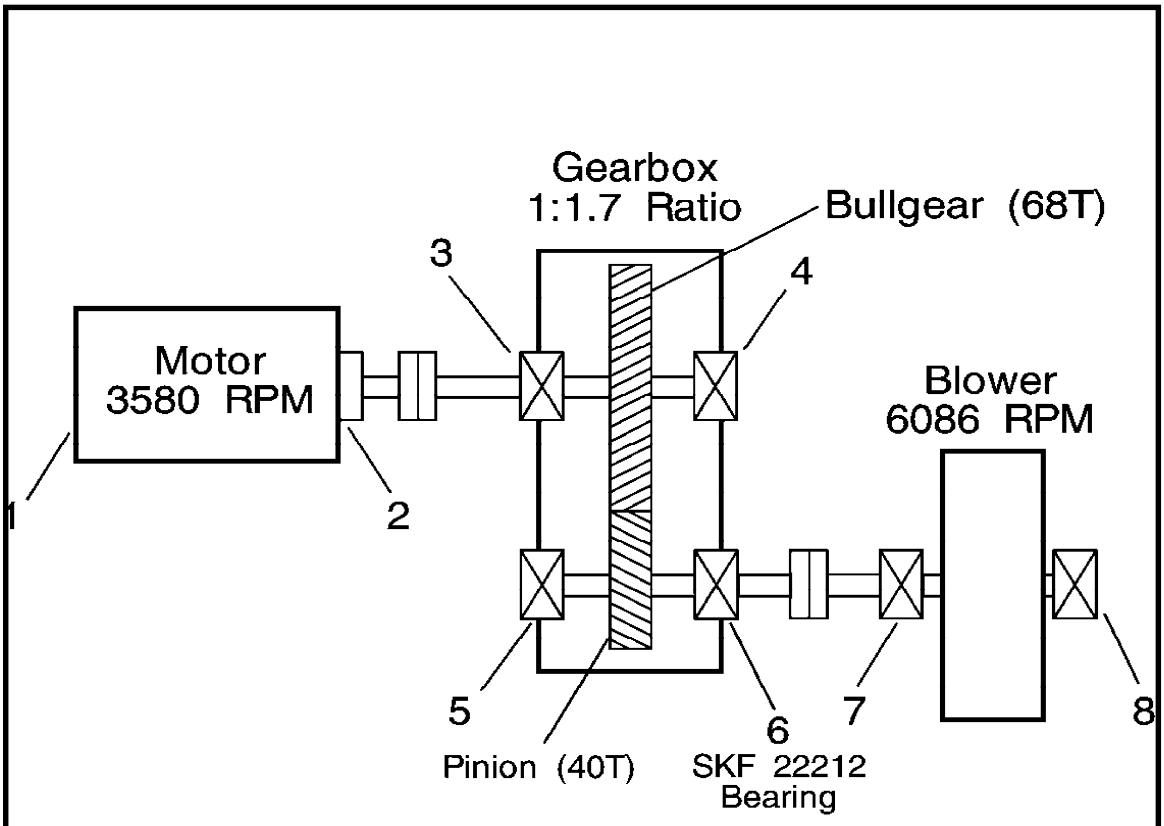
Figure 5 shows a spectrum collected on the PMP route with F_{MAX} set to 12,000 CPM. This frequency span would encompass as many as 3 harmonics of motor speed, but only the operating speed frequency of the output gear and blower (6086 RPM). Note that nothing unusual is present in this spectrum. The motor, blower, and 2X motor frequencies are present, but not at levels which would typically cause concern and this machine would be reported to be in "Good" or "Fair" condition. However, this is not the case as will be shown.

Figure 6 shows the same measurement point (position 6 of Figure 4) on the same machine as the plot in Figure 5, however, F_{MAX} has been increased to 24,000 CPM. It shows that harmonics of the gearbox output frequency are present which indicate possible machine faults (such as slight looseness, or possibly a bearing turning on a shaft). Also, it is important to point out that with the higher F_{MAX} setting, the resolution (ability to differentiate peaks) has decreased. Therefore, more than one measurement may need to be taken at this point - one with a low F_{MAX} to differentiate 2X motor frequency (7160 CPM) from 2X electrical line frequency (7200 CPM) and another with a higher F_{MAX} to pick up the gear and bearing fault frequencies.

Figure 7 shows another plot from gearbox position 6 as in Figures 5 and 6. However, F_{MAX} has been significantly increased to 240,000 CPM (up to 39.4X blower RPM). Note now that 5 harmonics of the gearbox bearing outer race frequency (BPFO) have appeared, along with the bearing fundamental train frequency (FTF) sidebands, none of which were even present in the previous spectra (note that FTF is often referred to as the "cage frequency of a bearing"). This signifies an extremely dangerous situation which should be corrected as soon as possible. This spectra shows the SKF 22212 gearbox bearing approaching catastrophic failure, but it would not have even been noticed had F_{MAX} not been noticeably increased.

Figure 8 shows the same measurement point on the same machine as Figures 5 thru 7. However, F_{MAX} has been increased to 360,000 CPM. Now, the fundamental gear mesh frequency (GMF) has appeared for the first time as we went to higher and higher frequencies. Also, note the output gear shaft speed sidebands spaced on either side of GMF. It is quite common to see 1X gear mesh frequency in a spectrum, and is not necessarily considered a problem. The bearing defect is still clearly the most dominant problem noted by this spectrum.

© Copyright 2001 Technical Associates Of Charlotte, P.C.



SKF 22212 BEARING DEFECT FREQUENCY MULTIPLES (CPM) AT 6086 RPM

DEFECT FREQ.	DEFECT COEF.	BEARING DEFECT FREQUENCY MULTIPLES				
		1X	2X	3X	4X	5X
BPFO	7.26X	44,184	88,368	132,552	176,736	220,920
BPFI	9.72X	59,156	118,312	177,468	236,624	295,780
FTF	.43X	2,617	5,234	7,851	10,468	13,085
BSF	3.33X	20,266	40,532	60,798	81,064	101,330

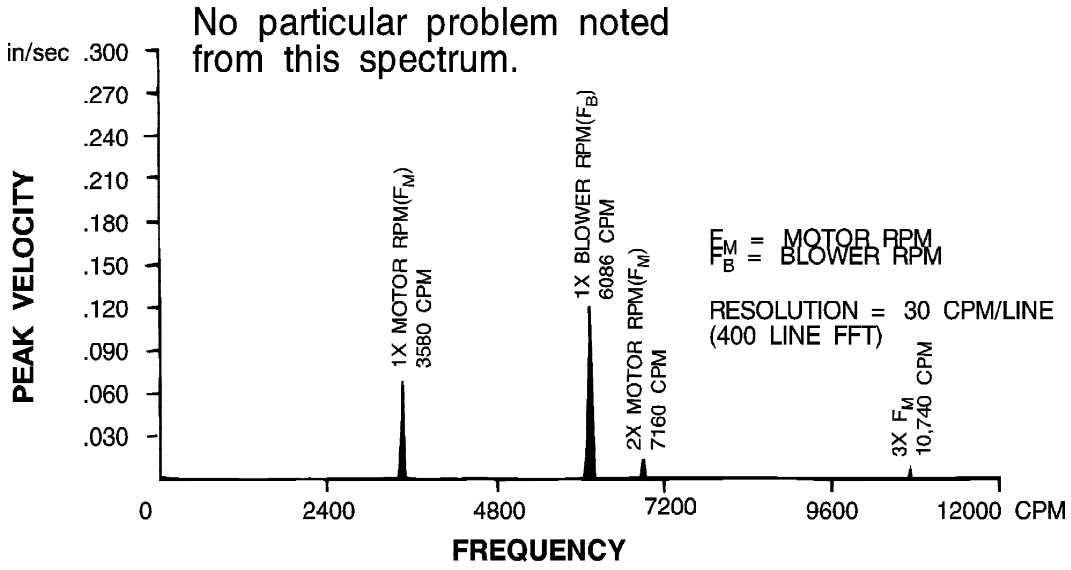
GEAR MESH FREQUENCY MULTIPLES

1X GMF	243,440 CPM
2X GMF	486,880 CPM
3X GMF	730,320 CPM

GMF = # TEETH X RPM
= 68T X 3580 = 40T X 6086
GMF = 243,440 CPM

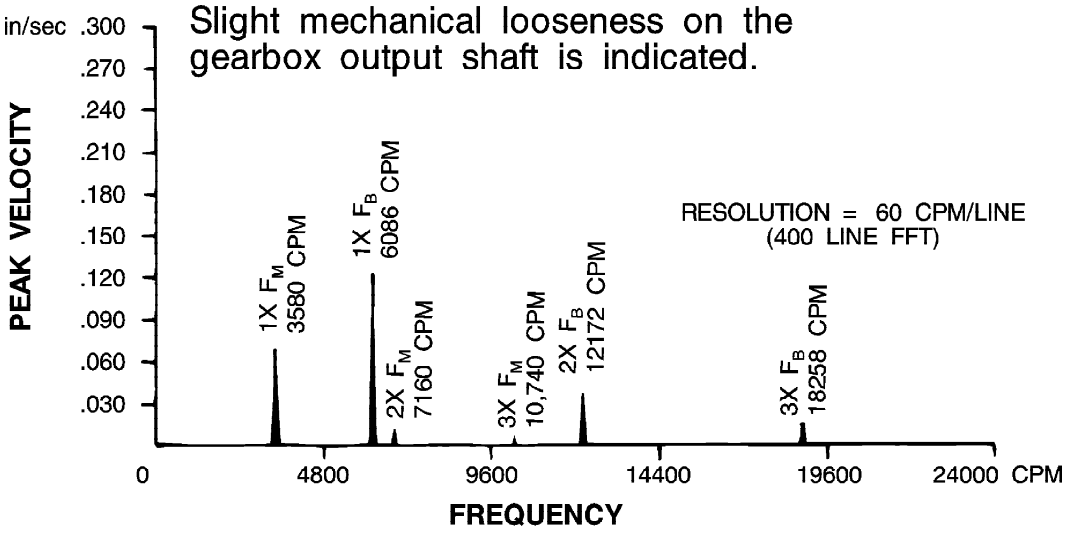
MOTOR-DRIVEN GEARBOX AND BLOWER MACHINE DRAWING
FIGURE 4

© Copyright 2001 Technical Associates Of Charlotte, P.C.



0-12,000 CPM SPECTRUM CAPTURES 1X THRU 3X GEARBOX INPUT SHAFT RPM, BUT ONLY 1X GEARBOX OUTPUT SHAFT RPM GOOD RESOLUTION OF 30 CPM/FFT LINE.

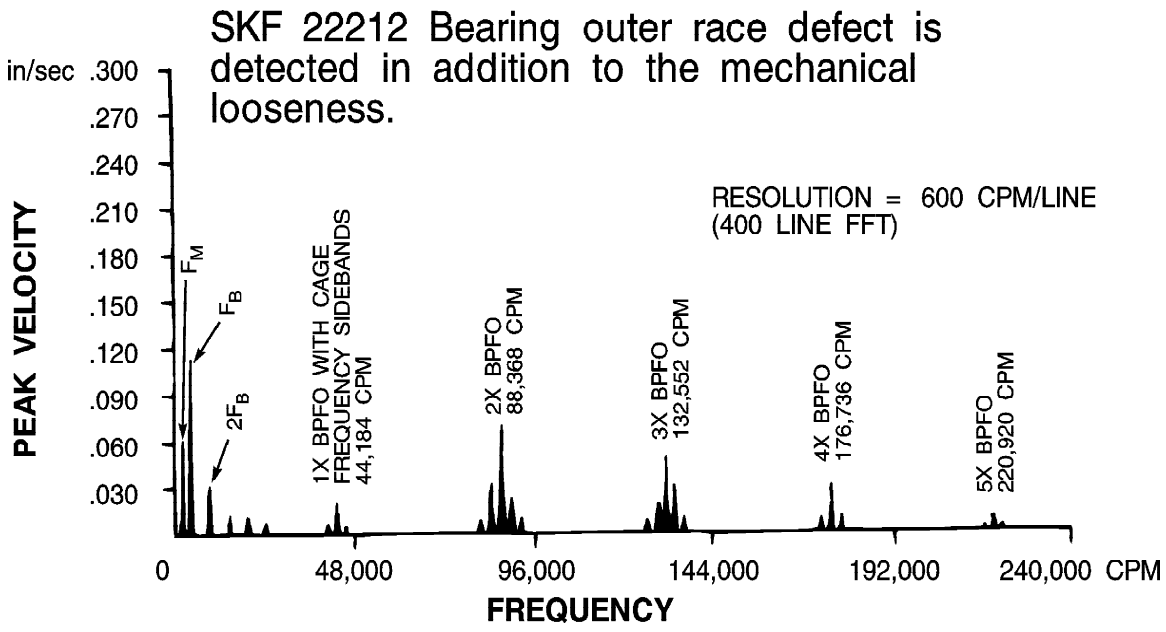
FIGURE 5



0-24,000 CPM SPECTRUM CAPTURES 3 HARMONICS OF GEARBOX OUTPUT SHAFT RPM IN ADDITION TO INFORMATION FROM FIGURE 5 ABOVE. RESOLUTION DECLINES SLIGHTLY.

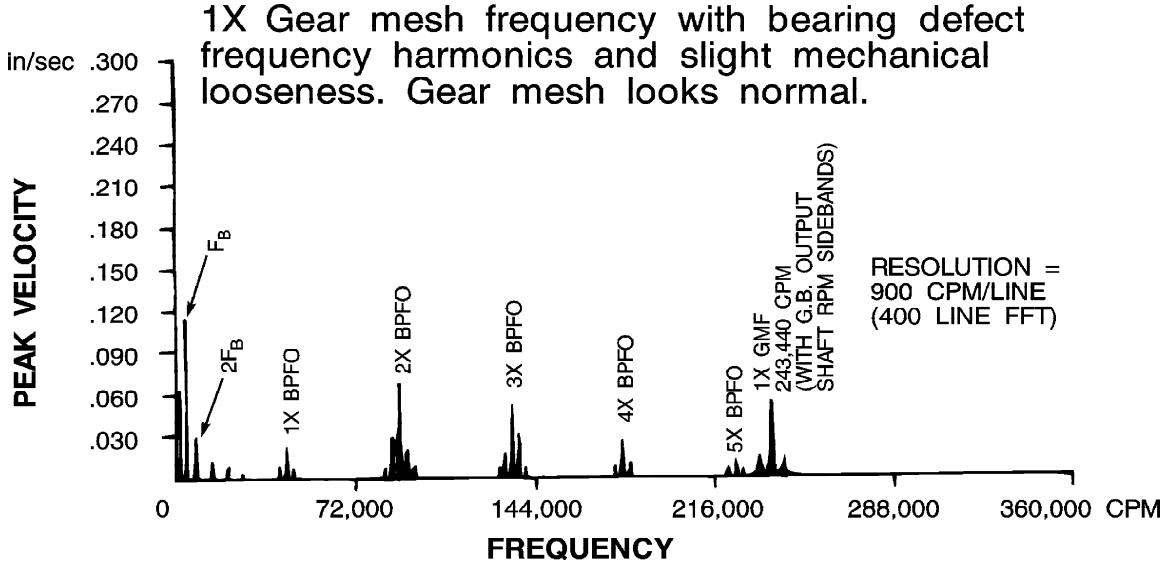
FIGURE 6

© Copyright 2001 Technical Associates Of Charlotte, P.C.



0-240,000 CPM SPECTRUM CAPTURES 5 PINION BEARING OUTER RACE DEFECT FREQUENCY HARMONICS IN ADDITION TO INFORMATION FROM FIGURE 6. RESOLUTION DECLINES TO 600 CPM/LINE.

FIGURE 7

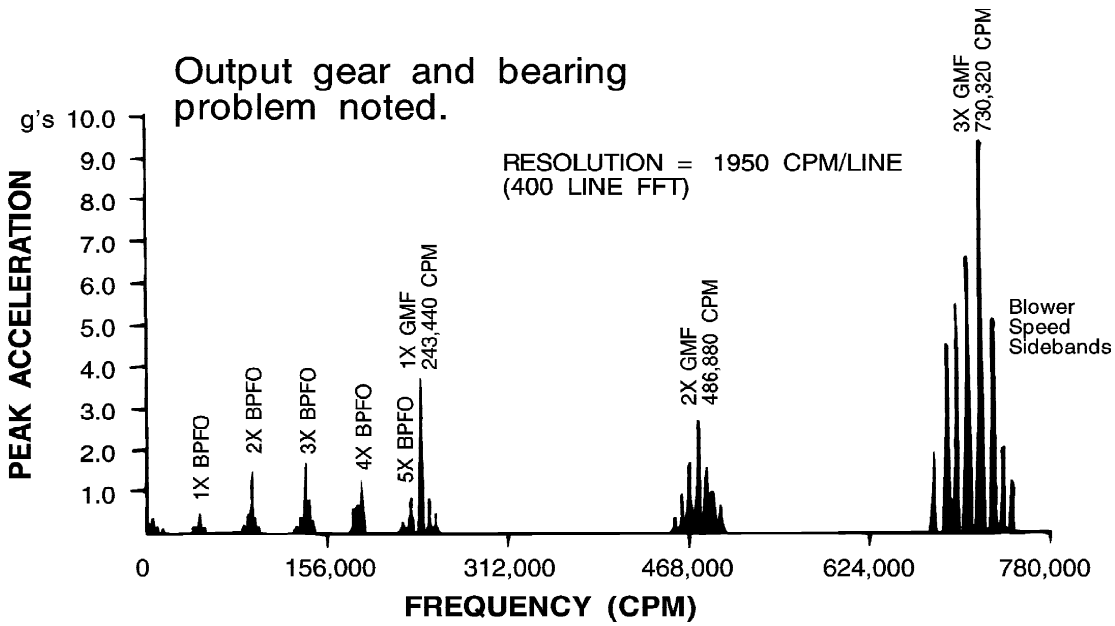


0-360,000 CPM SPECTRUM CAPTURES 1X GEAR MESH FREQUENCY IN ADDITION TO INFORMATION FROM FIGURE 6. RESOLUTION CONTINUES TO DECLINE.

FIGURE 8

© Copyright 2001 Technical Associates Of Charlotte, P.C.

Figure 9 shows data from the same measurement point as in Figures 5 thru 8, but F_{MAX} has again been increased up to 780,000 CPM. Also, note that the units have been changed to acceleration. This is because velocity response falls off in the higher frequency ranges where acceleration must be used. Note that we now see 2X GMF and 3X GMF. Also, note that 3X GMF is surrounded by a whole series of blower running speed sidebands which were not present in Figures 5 thru 8. In addition, note that 3X GMF is much higher than 1X GMF and dominates this acceleration spectrum. This machine must be attended to immediately with both a bearing and gear problem likely on the gearbox output shaft. However, it would not have been considered a serious problem machine if one had chosen an incorrect F_{MAX} such as that of Figures 5 and 6.



NOTE THE MEASUREMENT UNITS HAVE CHANGED FROM VELOCITY (in/sec) TO ACCELERATION (g's) DUE TO THE HIGH FREQUENCY RESOLUTION IS NOW SOMEWHAT POOR (1950 CPM), MEANING THAT 2 PEAKS MUST BE AT LEAST 2925 CPM APART FOR THEM TO EVEN BE DISPLAYED. A HIGH RESOLUTION SPECTRUM LIKELY USING 1600 LINES IS RECOMMENDED HERE. (Which would likely result in a resolution of 487.5 CPM - easily capable of displaying 1X RPM of motor and blower, along with clearly showing either bullgear (3580 RPM) or pinion speed (6086 RPM) sidebands if they are present).

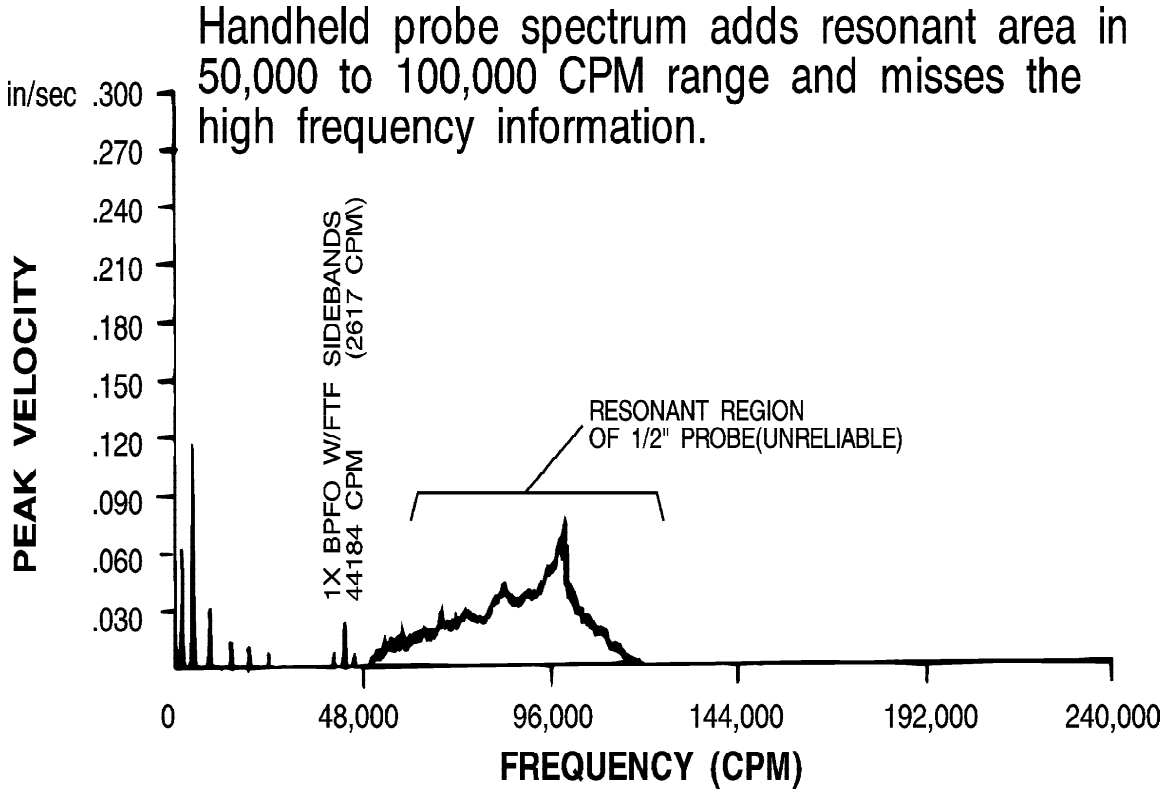
0-780,000 CPM SPECTRUM CAPTURES HARMONICS OF GEAR MESH FREQUENCY IN ADDITION TO INFORMATION FROM FIGURE 8.

FIGURE 9

© Copyright 2001 Technical Associates Of Charlotte, P.C.

8.14 INSTRUMENT SELECTION, SETUP, AND CONDITION

1) Consider the allowable frequency range for both the data collector, transducer, and transducer mounting for each PMP machine. For example, if taking data on a low-speed agitator with a motor speed of 1780 RPM, but an agitator speed of only 60 RPM, it will be necessary to use both a low frequency data collector and a low frequency transducer on agitator measurement locations. Furthermore, transducer frequency range (governed on the high end by natural frequency) must be considered. Consider Figure 10 where the machine monitored in Figure 4 was again measured with an accelerometer outfitted with a 1/2 inch handheld probe having a natural frequency of approximately 96,000 CPM. Note that only the data up to approximately 48,000 is considered reliable in this case (with longer probes, this reliable frequency limit drops even lower). Above this frequency, however, the probe is becoming resonant and actually contributing "false" vibration data in the range of approximately 50,000 to 100,000 CPM. In other words, the plot of Figure 7 shows this raised noise floor is not even present. Furthermore, once well above the probe mount natural frequency (over approximately 120,000 CPM for this 1/2 inch probe) there is little or no response to vibration that in fact does exist. Therefore, machine problems that are present at these higher frequencies, as shown in Figures 7 thru 9, are no longer reported resulting in a false sense of security and can lead to unexpected catastrophic failure. **Each probe and transducer has its own natural frequency and response curve which should be supplied when it is purchased. Be careful to fully understand exactly what that information is telling you.**



HAND-HELD PROBE SPECTRUM AS COMPARED TO FIGURE 7 SPECTRUM.

FIGURE 10

© Copyright 2001 Technical Associates Of Charlotte, P.C.

2) Ensure the data collector batteries are in good condition and properly charged before beginning the measurement route.

3) Make sure cables are in good operating condition. Check them out before beginning the collection route.

4) Check for correct date and time in the data collector before taking data. This will affect both the overall measurement trends and spectral waterfalls since the date and time are "tagged" to this data in most PMP data collector/software systems.

5) Make sure the proper transducer sensitivity is entered into the data collector (as well as into the PMP software from each point if the transducer sensitivity is requested in the software point setup).

6) If additional equipment such as a temperature probe, current probe, phototach, proximity probe, or pressure transducers are used, ensure that their operation is clearly understood and the vibration instrument is properly configured to accept the information from them and connected to the proper input of the data collector. Ensure that the proper sensitivity is entered for the transducer (mV/psi for a pressure transducer; mV/amp for a current transformer, etc.)

7) Insure the transducer and/or other accessories are in good condition and are attached properly to the data collector. Check the connecting cable for proper operation, not only before, but periodically throughout the measurement route.

8) Be aware of instrument error stackup that can be as low as -14.5% and as high as 31.5%. The following excerpt from an article by John Catlin of IRD Mechanalysis (ref. 1) explains this:

The following table and text is reprinted from that article which indicates the amount of error that can be introduced by each particular piece of instrumentation or action:

	Lowest Variation	***Highest Variation***
Analyzer	*±5%*	*±10%*
Transducer	*±5%*	*±10%*
Magnetic Holder	*±0.5%*	*±0.5%*
Tape Recorder	*±1%*	*±2%*
Cable (Negligible if proper precautions are taken)		
Calibration	*±1.5%*	*±4%*
Attachment	*±1.5%*	*±5%*

"NOTE: The variations caused by factors of "transducer position", machine "environment", and "steady state" operation are not included in the above list because of the difficulty in establishing "lowest" and "highest" variation limits.

If the variations listed above are combined to give total maximum variations, the "Lowest Variation" range is -13% to +14%, while the "Highest Variation" range is -28% to +35%. The addition of "transducer position", "environment", and "steady state" variations can cause these percentages to increase significantly.

From the above, three conclusions can be drawn which should be helpful in carrying out a predictive maintenance program.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

1) Do not attempt to make judgments on a machine's condition from small changes in amplitude which could readily be caused by variability of the instrumentation, transducer attachment, or normal variations associated with the environment and "steady state" operation of the machine.

2) Try to minimize the variability of the measurements by standardizing on one type measurement system and careful measurement (e.g., transducer attchment) procedures; but don't become so concerned with accuracy that the measurement procedure becomes excessively tedious and lengthy. Remember that small machinery defects generally continue to perform their required function even when balance levels increase by a factor of 2 to 10, or rolling-element bearing levels increase by a factor of 5 to 50. Thus, in most cases it is not necessary to make decisions on machine shutdown and repair based upon small changes in vibration amplitude.

3) Learn to recognize those machines whose vibration signatures have significant variations under "steady state" conditions because of the random nature of the inherent exciting vibrations. Pumps and fans which move liquids and gases are examples of machines which can fall in this category. In measuring such machines, avoid making decisions on machine condition based on changes which fall within the range of amplitude variations normally experienced during "steady state" operation. Alternately, make use of averaging to minimize the effects of the random variations on the measured amplitudes.

In summary, when carrying out a predictive maintenance program try to retain the same system components when making measurements. If it is necessary to change components (e.g. analyzer, transducer, etc.), run a check to determine the effect of such change on the total system response; and take any observed variations into account when interpreting the measurement results. Use careful measurement procedures to minimize variations caused by such sources; and be aware that many machines, even when running under "steady state" conditions, can have significant variations in their vibration signatures."

8.15 MEASUREMENT TECHNIQUES

1) Do not allow the transducer to rock back and forth, or to slip while taking measurements. If it should happen to do so, repeat the measurement (or the results can be very inaccurate and unreliable).

2) Apply firm continuous hand pressure to any handheld measurement.

3) Be aware that magnetism and radio frequency (RF) waves can affect data collectors and/or transducers. If measuring in areas so affected, take special precautions such that both the analyzer and transducer are not effected.

4) Do not oversize a magnet and/or transducer with respect to the measurement location. This may affect or even change the vibration response if they are too massive, or if the magnetic field is strong enough to affect operation of the machine.

5) For low frequency measurements below 300 CPM, make sure the cable is not swinging back and forth. This motion can sometimes actually become part of the overall level or even part of the spectrum, and yet it has nothing at all to do with the machine itself.

6) Allow the vibration signal some time to settle before storing the measurement (normally 5 or 6 seconds unless taking data with a special low frequency transducer). Again, not doing so can cause very erroneous overall level and spectral results.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

7) When at all possible, have vibration amplitude in the upper one-third of the measurement scale on those data collectors which allow the user to view the overall level before storing it and initiating the spectral measurement. In some data collectors, the full-scale amplitude of the spectrum will be set by the overall level. This can detrimentally affect the dynamic range if too high a full-scale amplitude is chosen (for example, if one chose a full-scale of 1.0 in/sec, and the amplitude of the highest peak was only .12 in/sec, the resulting spectrum may or may not pick up bearing defect frequencies with amplitudes of only about 0.010 to 0.015 in/sec).

8) Use the analysis and inspection features of data collectors if particular problems are noted and the additional information would be helpful. Examples of common inspection notes are: “leaking oil”, “hot bearing”, “loose belt”, “machine down”, “gravelly noise”, etc.

8.16 TRANSDUCER MOUNTING AND PROBES

1) Use a magnet or quick-lock mount for taking data wherever possible. A “quick-lock” mount is a permanently mounted mechanism which allows certain transducers to be connected to it for the measurement by only a quarter turn of the transducer. It usually produces even better results than does the magnet mount.

2) Do not use a probe unless absolutely necessary. Even short 1/2 inch long probes can produce very unreliable results particularly if real frequencies exist above approximately 45,000 to 60,000 (750 to 1000 Hz) as was shown in Section 8.13. **Figure 11** shows 3 different spectra taken from the identical point on an induction motor driving a centrifugal air compressor. The only difference was the transducer mount itself. This was a 2-pole motor running at approximately 3563 RPM. This motor was outfitted with 45 rotor bars. Therefore, the Rotor Bar Passing Frequency (RBPF) was approximately 160,335 CPM (45 bars x 3563 RPM = 160,355 CPM). Here, it is important to point out that the same accelerometer and analyzer were used with all 3 measurements taken within a five minute interval. Looking at the spectra of Figure 11, note that the “magnet mount” spectrum at the top of the figure easily detected this Rotor Bar Passing Frequency (RPBF) at about 160,312 CPM (importantly, note that RBPF was by sideband frequencies spaced at exactly 7200 CPM which is 2X electrical line frequency). This is a common phenomenon in that rotor bar looseness is characterized by a high amplitude RBPF surrounded by 2X line frequency sidebands. On the other hand, the handheld measurement in the middle of the figure nearly missed RBPF altogether. And, even worse, when the 9" probe was attached to the accelerometer and the measurement repeated, not only did the probe mount totally miss the RBPF problem, but it also greatly amplified frequencies in the region of 40,000 to 60,000 CPM. Therefore, this figure is proof that a handheld probe should only be used when absolutely necessary since it will not only greatly amplify frequencies in one region, but will entirely miss important high frequency problems altogether.

Another example of the unreliability of probes or handheld mounts is shown in **Figure 12**. In this figure, an IRD 970 accelerometer was used with four different mounting techniques:

A) Glue

B) Magnet

C) Handheld Only

D) 9" Probe

Figure 12 - (A) is the most reliable data since it was collected with an adhesive mount. As can be seen, there is a bearing problem since the fundamental bearing defect and its harmonics are present.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

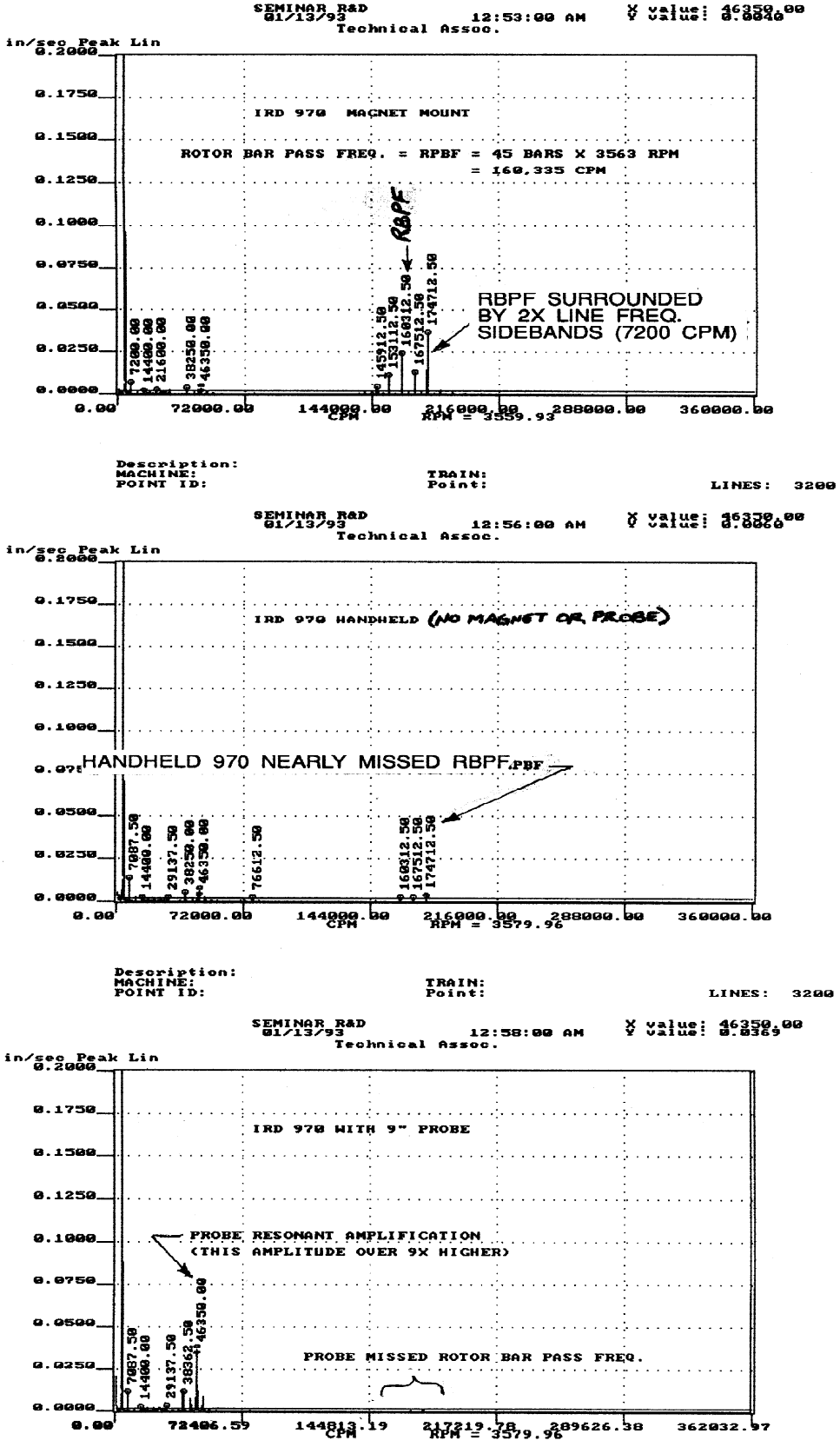


FIGURE 11
EFFECT OF TRANSDUCER MOUNT
ON RESULTING VIBRATION SPECTRUM

© Copyright 2001 Technical Associates Of Charlotte, P.C.

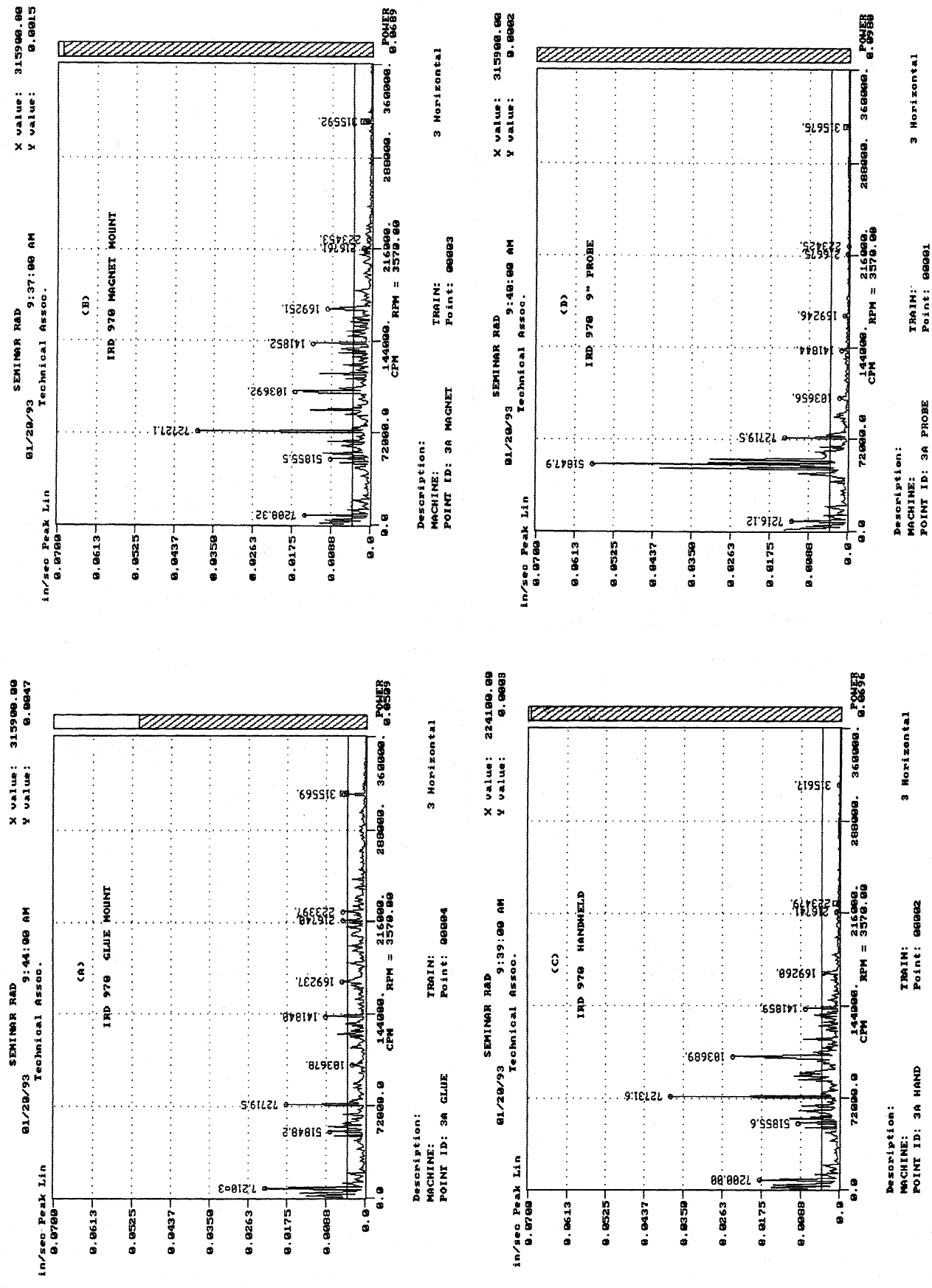


FIGURE 12

© Copyright 2001 Technical Associates Of Charlotte, P.C.

Figure 12 - (B) is the next best data since it was collected with a magnet mount. However, it is becoming more difficult to see the frequencies over 200,000 CPM since the magnet mount is not sufficient to “pick-up” those higher frequencies.

Figure 12 - (C) was collected with a handheld mount (no probe or magnet) and it is unacceptable. The bearing frequency harmonics no longer show up as peaks and this makes diagnosis on the condition of this bearing impossible. These peaks will never appear, even though they are present as can be seen in **Figure 12 - (A)**, since a handheld accelerometer is incapable of recording these high frequencies.

Figure 12 - (D) is the worst of all since it uses a 9" probe mount. This mounting procedure is unacceptable also. Not only are bearing frequencies missing but, the frequencies in the range of 40,000 to 60,000 CPM have been amplified greatly due to the resonance of the 9" probe. The 9" probe is only recommended in areas that are inaccessible or unsafe. These points should have permanent accelerometers mounted instead of using the probe.

3) Do not take measurements on curved surfaces without using special precautions or a barred magnet. Doing so can greatly reduce the effective accurate frequency range of the transducer/magnet system.

4) Ensure contacting surfaces between the magnet and transducer are tight and free of dirt, grease, or other contamination. In addition, it likely will be necessary to scrape off the paint from the measurement location area if frequencies exceeding approximately 2000 Hz (120,000 CPM) are to be measured. Not doing so can significantly dampen or even eliminate the response from these high frequencies.

5) For the same reason as in Item 4 of this section, if mounting permanent washers or discs to a machine to provide a good measurement surface for the transducer, keep the thickness of the adhesive thin to allow good transmission of high frequencies through the adhesive itself. Also, ensure that the adhesive used has been tested and proven capable of freely passing these high frequencies (some adhesives themselves noticeably dampen such signals). Finally, if the permanent washer surface is uneven or rough, make sure that an adhesive is used which has the capability of filling these voids to provide a solid path for high frequency signal sources.

6) Ensure the transducer and/or magnet are held against a clean, dirt-free surface when taking measurements.

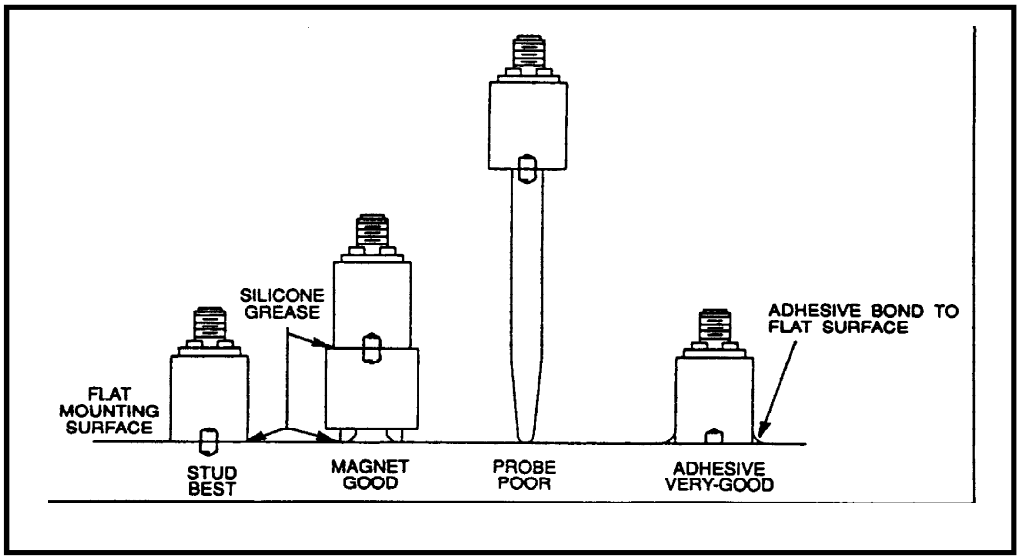
Pickup Mounting Methods

It is especially important that you meet certain requirements for ensuring the accuracy and repeatability of Spike Energy measurements. The technique you use for mounting the pickup is particularly important in this regard. Illustration A illustrates various mounting methods.

Stud mounting (or attachment with a high modulus adhesive bond) provides an effective method to attach an accelerometer to the machine surface. The primary advantage of this technique is that it provides the best path for the transmission of low level, high-frequency vibration from the machine to the pickup. It is, however, generally not suitable for periodic checks because attachment cannot be made quickly.

Another method uses a magnetic holder. Although a magnetic holder is simpler to use for periodic checks, there will be some loss of vibration energy between the machine and accelerometer. You can thus expect lower Spike Energy measurements than you would find with a stud-mounted accelerometer in the same position. For repeatability, care must be taken to locate the magnetic holder at precisely the same position each time measurements are made.

© Copyright 2001 Technical Associates Of Charlotte, P.C.



MOUNTING METHODS FOR SPIKE ENERGY PICKUPS
ILLUSTRATION - A

The third method uses a hand-held probe. The Spike Energy readings are normally lower than you would find for any of the other mounting techniques. This is mainly due to the attenuation of the probe. Again, repeatability of readings on a periodic-check program is dependent on the consistency with which you place the probe. The hand-held probe has the advantage that it can often be located closer to the bearing (tip contact) than when using the other mounting methods.

NOTE:

Do not compare Spike Energy measurements from the same point unless they are taken under the same conditions, including the mounting method.

It is important that you attach the accelerometer in exactly the same place each time a measurement is made. A circle painted around a mounted accelerometer makes it easier to reattach it in the proper position for future measurements.

The surface where the measurement is to be taken must be kept clean and bare. Surface materials such as paint or grease can act as a spring under the accelerometer, thus lowering its resonant frequency and reducing high-frequency Spike Energy amplitudes. Also, the surface should be flat so that the accelerometer does not wobble when attached. When using a stud mounting, its axis must be perpendicular to the surface so that the accelerometer sets squarely on top of it.

Use a light coating of silicon grease or lubricating oil between the accelerometer and mounting surface. This forms a stiff, incompressible layer that transmits high-frequency vibration. When using a magnetic holder, applying a light coating of oil to the mounting surface improves high-frequency transmission.

8.2 EFFECT OF TRANSDUCER MOUNTING ON VIBRATION MEASUREMENTS

The following is an article by CSI (ref. 2) which outlines the effects of transducer mounting on the usable frequency range. The transducer mountings studied are stud, adhesive, quick-lock and magnetic. It is considered one of the better studies on this important topic published to date. Note particularly what is said about each of the mount configurations (esp. the information on hand-held probes versus magnets or quick-lock mounts). The article also provides good information on what adhesives should be used when mounting disks or washers permanently.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

Real-World Mounting of Accelerometers for Machinery Monitoring

S. V. Bowers, K. R. Piety and R. W. Piety, Computational Systems Incorporated, Knoxville, Tennessee

Typical accelerometer mounting configurations were evaluated to determine acceptable frequency response and resonant frequencies for each mounting. Basic configurations studied included stud, adhesive, quick lock and magnetic. A hand held probe with varying stinger lengths was also evaluated. In addition, the effects of bad mounting techniques were considered. Frequency ranges and resonances were determined for sensor mounts exposed to rocking, looseness and varying hand pressure.

Predictive maintenance programs that operate in utilities and major industries rely heavily on manual data collection. The temporary sensor mounting techniques utilized in manual data acquisition significantly impact data repeatability and reliability and thus the decisions based on these data. The benefits of these monitoring programs are dependent upon accurate data collection techniques which affect both detection and diagnosis. Provided vibration instrumentation is calibrated, accurate vibration measurements are ensured when: (1) sensor mountings do not limit frequency and dynamic ranges; (2) sensor mass does not change the vibration characteristics of the test object; and (3) measurement locations are exactly repeated.[1]

The objective of this project was to evaluate the limitations of some common temporary sensor mounting techniques as they apply to predictive maintenance programs. This was achieved in three major sets of experiments. For the first set, maximum frequency responses were determined for common sensor mount configurations. In the second set of tests, misapplication of sensor mounts was investigated. The third set of experiments explored the effects of several sources of variability on vibration measurements.

Frequency response limits for a given sensor are defined as the linear range of the frequency response curve for that sensor. It is desirable for the output from the sensor to be linearly related to the input over as wide a frequency range and dynamic range as possible.[1] A main focus of this study was to determine sensor frequency response limits resulting from various mount configurations. In practical applications, the actual frequency response of a sensor depends upon the dynamic response of the sensor/mount configuration. Vibrations at frequencies near the sensor/mount configuration resonant frequency will be amplified, whereas at other frequencies the measured vibration is greatly attenuated. The frequency ranges and resonant frequencies of various mount fixtures are reported in this article.

The usable frequency range of a mounted sensor can be reduced by rocking and loose connections. In addition to these factors, the frequency range for hand held probes can be reduced due to variations in hand pressure. Limitations imposed by these misapplications were evaluated. Because of improper mounting and irregular or dirty surfaces, rocking occurs, which tends to decrease the usable frequency range of the transducer.[2] Looseness occurs when mount configurations and sensor are not tightly connected. This problem results in large errors at high frequencies,[3] as well as chattering. Hand held probes can also experience reductions in frequency response due to chattering because of improper hand pressure.[3] Any action that allows the pickup axis to change direction also produces incorrect readings.

Based on a paper presented at the 4th Incipient Failure Detection Conference, Electric Power Research Institute, Philadelphia, PA, October, 1990.

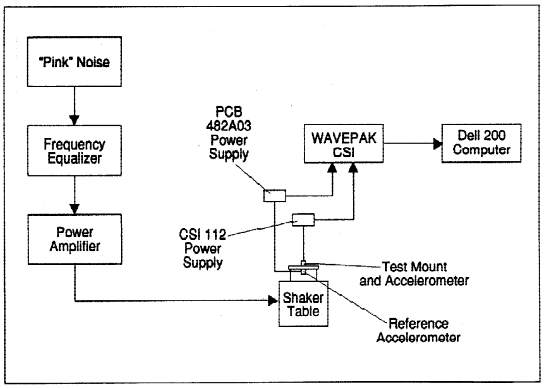


Figure 1. Schematic of instrumentation to evaluate sensor mount configurations.

Other sources contributing to variability in measured results are the mount positions on vibrating surfaces, varying sensor placement techniques by different people, and individual consistency. This article presents results relating to these factors and attempts to quantify their effects on measured values in typical or extreme cases. Tests were evaluated on industrial machines from data obtained with an accelerometer and a hand held probe using common mount configurations currently employed in industrial predictive maintenance programs.

Sensor Mount Configurations

During the sensor response tests, accelerometer outputs were measured using a CSI Wavepak which is a two-channel FFT spectrum analyzer integrated with a personal computer. Each measurement, unless otherwise noted, was taken with 800 lines of resolution and 200 averages. The instrumentation employed in this study is indicated in the block diagram of Figure 1. "Pink" noise was generated and passed through a frequency equalizer to provide a relatively constant output per unit frequency between 0 and 20,000 Hz. Output of the power amplifier was adjusted to insure an overall RMS value of 1 g in the measured spectra.

Test data were comprised of transfer function, phase and coherence curves determined from test and reference accelerometer measurements of each mount configuration. The reference accelerometer (Wilcoxon 726) was always stud mounted on the bottom of the plate and torqued to 20 in-lbs. All test accelerometer measurements were taken from the top of the plate. With the exception of the hand held accelerometer tests, experiments were conducted using a Wilcoxon 726T accelerometer as the test accelerometer. The manufacturer rated the frequency response of the 726 accelerometer at 3 to 10,000 Hz (±5%). When stud mounted, the resonant frequency for this accelerometer is 35,000 Hz. The rated frequency response of the CSI 310 hand held probe stud mounted and tightened to a vibrating surface by hand is 2 to 2000 Hz.

Test configuration data were normalized by base data in order to reduce or remove measurement bias resulting from the test procedures employed. Normalization requires complex division of the test data by the base data. The normalizing process was considered valid provided both the test and base data had a coherence of 0.95 or above. Base data were taken

© Copyright 2001 Technical Associates Of Charlotte, P.C.

periodically throughout the study to verify the consistency of test conditions and results. The base data were acquired with the test accelerometer (Wilcoxon 726T) stud mounted to the top of the steel test fixture plate to a torque of 15 in-lbs and the reference accelerometer stud mounted to the bottom of this test fixture plate to a torque of 20 in-lbs.

By normalizing the test data, a transfer function ratio, minimum coherence curve and phase difference were determined. Any transfer function ratio deviation exceeding 0.90 and 1.10 (greater than ±10%) was considered significant as was a phase difference greater than ±10°. These limits were selected for the purpose of establishing a recommended linear range of application for each specific mount configuration. Resonant frequencies of each mount configuration were also determined when possible. The resonant frequency of a configuration was noted when a transfer function ratio peak occurred at a phase difference of approximately 90°.

Results, Discussion and Conclusions. Basic mount configurations considered were: 1. stud mount, 2. adhesive mount, 3. quick-lock mount, 4. magnet mount, and 5. hand held probe. For the most consistent and reliable data, it is best that readings from any given measurement point be taken from smooth surfaces where the path from the sensor to the monitored point is direct. This is best accomplished by stud mounting a steel disk at desired measurement points. The only requirement for stud mounting a disk to a machine is that the machine surface be smooth and flat. A 1 in. diameter, 3/8 in. thick steel disk was stud mounted to the vibrating surface in this study. The frequency range of the accelerometer stud mounted to this fixture was 3 to 16,125 Hz as dictated by the phase difference boundary. This range is easily large enough for practical applications. For cases where the mounting surface is round, the disk may be applied using certain cement adhesives. Evaluation of a good adhesive for such a use will be discussed shortly.

When mounting disks, or any sensor mount, it is important that the machine surface be smooth especially when higher frequency measurements are desired. Voids between the measured surface and the sensor will reduce the accelerometer's frequency range. This is best demonstrated by comparing experiments conducted with and without wax. After applying wax to a fixture, frequency ranges were enhanced by as little as 925 Hz to as much as 4,750 Hz. The frequency ranges were improved because voids between the mount fixture and vibrating surfaces were filled allowing better transmission between contacting surfaces. In practical measurement situations, applying wax is not feasible due to surrounding debris and excessive temperatures. However in situations where higher frequencies must be analyzed, wax can be applied provided the environment is relatively clean and the temperature is below 200° F (wax melts above 200° F).

As alluded to previously, steel disks can also be bonded to machines via adhesives. The only requirements are that the surface be clean and smooth (no drills or taps needed). Mounting procedures for the tests in this study were identical to those required in industrial applications. Each adhesive was applied using the procedures described by the manufacturer of the particular bonding agent.

The accelerometer's frequency range resulting from the configuration using the Hottinger Baldwin Messtechnik X60 adhesive is 3 to 8,775 Hz. When the setup used the Loctite Black Max cyanoacrylate ester adhesive, the accelerometer's frequency range was 3 to 10,975 Hz. No resonance was observed for either of the adhesive mount configurations. Of the two bonding agents, Loctite Black Max was easier to apply. However, this adhesive will only bond surfaces that are flat and smooth. In situations where one or both of the mating surfaces is rounded (not flat), Hottinger Baldwin is ideal.

Sometimes, sensors must be isolated from mount configurations. In such situations, electrical insulators are placed between the sensor and mounting interface. The use of insulators tends to lower the maximum acceptable frequency. Three common insulators were evaluated in this study. The maximum acceptable frequency of the stud mounted accelerometer with insulators ranged from 4200 to 5400 Hz. The transfer function ratios and phase difference curves for these three tests are plotted in Figure 2.

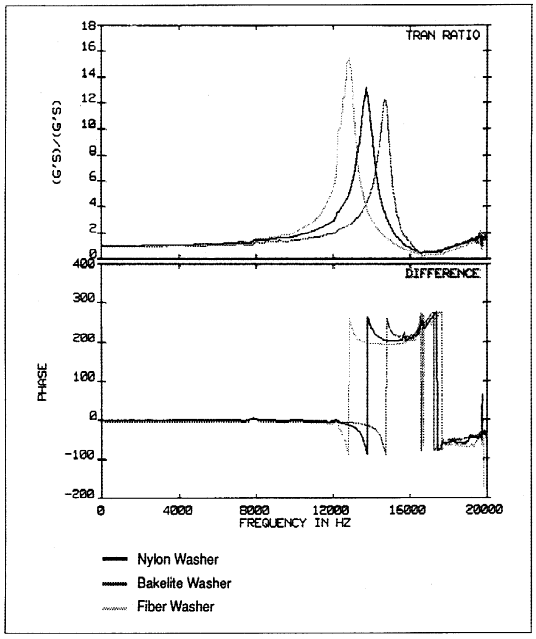


Figure 2. Transfer function ratios (top graph) and phase relationships (bottom graph) resulting from stud mounting of a Wilcoxon 726T accelerometer with three different insulated washers.

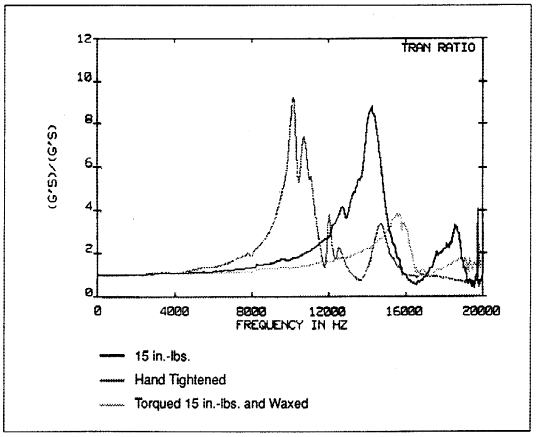


Figure 3. Transfer function ratios for three different quick lock configurations.

One mounting system used in lieu of the steel disk is the quick lock. This system consists of a base permanently attached to a measurement point and a cap connected to the sensor. The cap is easily coupled to the base with 3/4 of a turn. This connection is usually tightened by hand. Results of a test where the base was stud mounted at 15 in-lbs to the vibrating surface and the remainder of the quick lock system and sensor were hand tightened to the base show an acceptable frequency response ranging from 3 to about 4,000 Hz (see Figure 3). While a test was not conducted with the quick lock base cemented to a vibrating surface, the expected frequency response of such a configuration would be the same as the stud mounted quick lock system, i.e. 3 to 4,200 Hz. The frequency range of the quick lock configuration is limited because of the quick lock device, not the method of permanent connection to the vibration surface.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

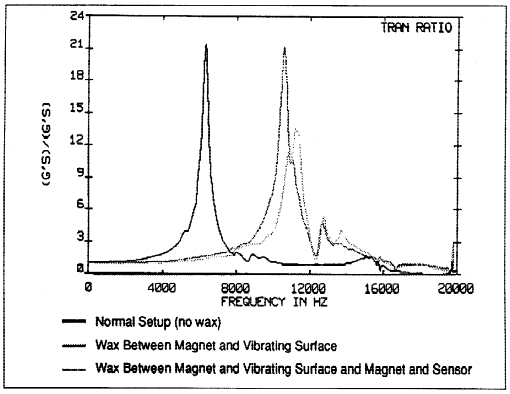


Figure 4. Transfer function ratios for three configurations of a Wilcoxon 726T accelerometer mounted on a Balmac magnet.

In many situations, permanent vibration mounting disks are not feasible because of accessibility and location of measurement points. The best solution in this case is to mount sensors on magnets. Three different magnets were evaluated in the experiments. For each mount, the accelerometer was stud mounted at 15 in-lbs to the magnet being tested. The frequency range of the accelerometer mounted on a rare earth magnet was the widest and ranged from 3 to 3,350 Hz. The frequency response of the Balmac magnet and CSI 906 super magnet were similar with ranges from 3 to 1,775 and 3 to 1,475 Hz, respectively. An example of transfer function ratios from the Balmac magnet configurations with and without wax are plotted in Figure 4. Even though the rare earth magnet had a better response, this magnet can only be used on flat surfaces because the face of the magnet is flat. The Balmac magnet is capable of holding onto flat and slightly rounded surfaces, while the design of the CSI 906 super magnet allows it to be used on practically any rounded or flat surface.

The final mount technique studied utilized a CSI 310 hand held probe and stinger configuration. Hand held probes should be used when measurement points are inaccessible by any other method. Typical stinger lengths are 2 and 8.5 inches. The maximum acceptable frequency for these configurations was much less than any of the other tests. For the hand held probe with a 2 in. stinger attachment, the maximum acceptable frequency was 525 Hz and for the 8.5 in. stinger attachment it was 350 Hz. It is not recommended that hand held probes be mounted on magnets because cantilever action can occur which will produce erroneous data.

The most common mount configurations of those evaluated are ranked in Table 1 with respect to the valid frequency response in decreasing order. The suggested frequency ranges and resonant frequencies are liberal estimates and will differ slightly under various conditions and for different sensors. The value observed for the recommended frequency range was influenced by the shape of the transfer function ratio curve for each mount. If the curve was smooth and sloping gently upward, the recommended value was chosen objectively at the base of the resonant curve where the graph was generally flat (linear relationship). Therefore, it is possible for the recommended maximum value to be larger than that observed.

Table 1 also shows that different mounting configurations tend to amplify measurements about their resonant frequencies to varying degrees. In general, hand held probe measurements amplify signals about the resonant frequency less (6×) than do the other configurations. One can expect a mounting system employing insulators to increase readings about the resonant frequency by about 12 to 16× the actual vibration. Magnetically mounted sensors will amplify vibration around the resonant frequency by 13× when using a rare earth magnet to over 20× when using a CSI 906 or Balmac magnet. Finally, when using a quick lock configuration, vibration can be amplified about the resonant frequency by 10×. These data suggest that measurements should not be taken at or around resonance when actual amplitudes are desired because the true vibration magnitudes will be amplified.

Poor Sensor Mounting Practices

Various misapplications associated with sensor mounting were investigated. The equipment and procedures used for tests in the previous section were applied to the experiments in this section. A description of the mount configurations tested is outlined in Tables 2 through 4. The maximum acceptable frequency with respect to the transfer function ratio and phase difference along with the resonant frequency for each mount are also recorded in Tables 2 through 4.

Results, Discussion and Conclusions. In most "real world" applications, the mounting configuration can be far from ideal. For example, rocking occurs when the vibrating surface and the mount or sensor do not fully and completely mate. Undesirable debris accumulation between the two surfaces tends to rock the sensor. To evaluate the severity of this problem, ideal and rocking modes were measured for stud mount and magnet mount configurations. Rocking was induced by randomly placing drill press residue between the test fixture plate and the contacting surface of several mounting configurations. Extra rocking was introduced into the magnetic mount by piling metal shavings in a row which created a "seesaw" effect.

The effect rocking has on a sensor's frequency range was apparent. Without the presence of rocking, earlier results revealed the frequency range for a basic stud mounted accelerometer to be 3 to 17,450 Hz. With rocking, the frequency

Table 1. Maximum acceptable measurement frequency limits and mounting resonances.

Sensor Mount Configuration*	Maximum Acceptable Frequency (Hz)**	Mounting Resonance Freq(Hz)	Mounting Resonance Ampl(g)
Sensor stud mounted to 3/8" steel plate stud mounted to vibrating surface	16,200	>20,000	N.O.
Sensor stud mounted to steel disk bonded to vibrating surface with Loctite Black Max cyanoacrylate ester	10,975	N.O.	N.O.
Sensor stud mounted to steel disk bonded to vibrating surface with Hottinger Baldwin Messtechnik X60	9,000	N.O.	N.O.
Sensor stud mounted with a:			
bakelite insulator	10,000	14,650	12.0
nylon insulator	9,000	13,750	13.5
fibre insulator	9,000	12,075	15.5
Sensor stud mounted to rare earth magnet	7,500	12,075	13.0
Sensor mounted to quick connect stud mounted to vibrating surface (hand tightened)	6,000	10,150	9.0
Sensor stud mounted to Balmac magnet	3,000	6,325	21.0
Sensor stud mounted to CSI 906 super magnet	2,000	5,250	22.0
CSI 310 probe using a pointed 2 in. steel stinger†	800	1,475	6.5
CSI 310 probe using a blunt 8.5 in. aluminum stinger†	500	1,075	6.0

* All stud mounts torqued at 15 in-lbs unless otherwise noted.

** Maximum acceptable frequency chosen objectively at the base of the transfer function curve where the slope of the line begins to increase towards the resonant peak but is still generally flat (linear relationship).

Freq Mounted resonance frequency.

Ampl Normalized amplitude at the mounted resonance frequency.

N.O. None Observed

† Stingers tightened to accelerometer at 15 in-lbs. Medium pressure applied to hand held probe.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

ranges observed were 3 to 6000 Hz for the sensor torqued to 5 in-lbs, 3 to 2150 Hz for the sensor finger tightened and only 3 to 775 Hz when the sensor was loose (Table 2). These results showed reductions in frequency ranges of 11,450, 15,300 and 16,675 Hz, respectively. Degradation in frequency response can be partially attributed to loose sensor mounting.

Tests conducted with magnet mounts revealed reductions in frequency responses due to rocking as well. The effects of rocking reduced the maximum acceptable frequencies from as little as 500 Hz to as much as 1,775 Hz. An example showing the effects of rocking on a magnet mount is seen in the transfer function ratios of Figure 5.

Table 2. Evaluation of rocking sensor mounts.

Sensor Mount Configuration	Maximum Acceptable Frequency (Hz)	Mounting Resonance Frequency (Hz)
Metal shavings between contacting surfaces of the sensor, stud mounted to the vibration pad for:		
normal mount*	17,450	>20,000
5 in-lbs torque	6,000	14,350
finger tightness	2,150	7,350
loose connection	775	2,725
Sensor stud mounted to rare earth magnet:		
normal mount*	2,975	8,875
metal shavings between magnet and vibration pad	1,800	6,125
metal shavings piled to one side	1,200	5,075
Sensor stud mounted to Balmac magnet:		
normal mount*	2,400	8,525
metal shavings between magnet and vibration pad	1,200	4,000
Sensor stud mounted to CSI 906 super magnet:		
normal mount*	1,250	3,550
metal shavings between magnet and vibration pad	750	2,650

*Torqued at 15 in-lbs.

Table 3. Evaluation of loose sensor mounts.

Sensor Mount Configuration	Maximum Acceptable Frequency (Hz)	Mounting Resonance Frequency (Hz)
Sensor stud mounted to the vibration pad for:		
normal mount*	17,450	>20,000
loose mount	1,050	2,450
Sensor stud mounted to rare earth magnet:		
normal mount*	2,975	8,875
finger tight	2,200	6,725
Sensor stud mounted to Balmac magnet:		
normal mount*	2,400	8,525
loose mount	500	3,375
CSI 310 handheld probe using a pointed 2 in. steel stinger:		
medium pressure	425	1,210
loose stinger	385	1,045

*Torqued at 15 in-lbs.

Table 4. Results of hand pressure applied to a hand held probe.

Sensor Mount Configuration	Maximum Acceptable Frequency (Hz)	Mounting Resonance Frequency (Hz)
CSI 310 hand held probe using a 0.25 in. stud as a stinger at pressures:		
light	395	1,050
medium	550	1,420
heavy	605	1,630
CSI 310 handheld probe using an 8.5 in. aluminum stinger at pressures:		
light	280	750
medium	320	875
heavy	415	1,070

*Torqued at 15 in.-lbs.

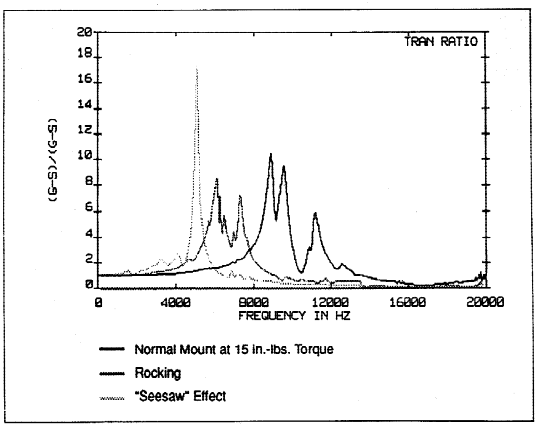


Figure 5. Comparisons with respect to the transfer function ratios of a Wilcoxon 726T accelerometer mounted on a rare earth magnet in the normal, rocking and seesaw modes.

Results suggest that measurement point surfaces and sensor interfaces be thoroughly cleaned and connections tightened in order to minimize the influence interface dynamics can have on data, particularly when data are being taken above 500 Hz. Furthermore, in a situation where a magnet is walking around on a surface or intermittently losing contact, low frequency motion and impacts (which are not generated inside the machine but due to a poor interface) will raise the noise floor of measured vibration spectra.

Mounting looseness occurs when any connection between the sensor and a vibrating machine is not securely tightened. Looseness often appears after moving from measurement point to measurement point without periodically checking the sensor and mount connection. Hand contact, vibrations from the equipment being monitored, and temperature fluctuations can eventually loosen sensors from their mounts. Looseness allows the sensor to vibrate randomly. This reduces the usable frequency range of the sensor and increases the probability of acquiring spurious data.

Two degrees of looseness were evaluated: "finger tight" and "loose mount." The finger tight mount was achieved by twisting the accelerometer to the mount with the fingers only. Thus, the sensor was snugly mounted, but not to the degree of hand tightness. The degree of looseness was defined as "loose mount" when movement between the sensor and the mount could be described as a random motion.

The usable frequency range measured for the "loosely" stud mounted accelerometer is 3 to 1050 Hz (Table 3), a reduction of 16,400 Hz from the basic stud mount setup. In the experiment where the sensor on a rare earth magnet was tightened with only the fingers, the frequency range was reduced by 775 Hz so that the adjusted frequency range was 3 to 2200 Hz. The frequency range with the accelerometer "loosely mounted" on a Balmac magnet was 4 to 500 Hz, a reduction of 1900 Hz to the regular Balmac magnet mount frequency range. The effect of this loose mount setup was further observed by evaluating the transfer function ratio and phase difference curve of Figure 6. The small peaks appearing at various frequencies indicate that the sensor was chattering. Because of chattering, high erroneous readings can be obtained. Such readings, especially at low frequencies, could mislead the analyst to conclude that mechanical problems are present.

With regard to the hand held probe, a loose connection produced the sensor frequency range of 3 to 385 Hz, a reduction in frequency response of 40 Hz relative to the preferred hand held

© Copyright 2001 Technical Associates Of Charlotte, P.C.

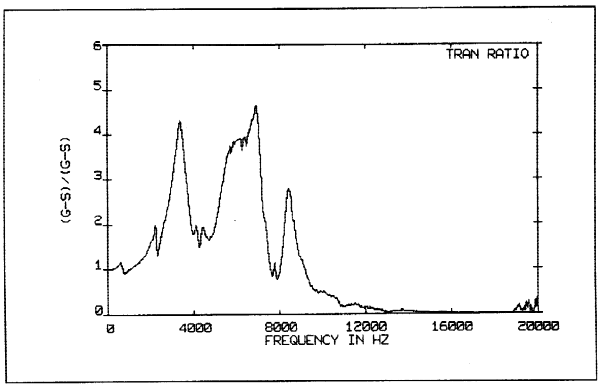


Figure 6. Transfer function ratio showing chattering produced by a loose connection between a Wilcoxon 726T accelerometer and a Balmac magnet.

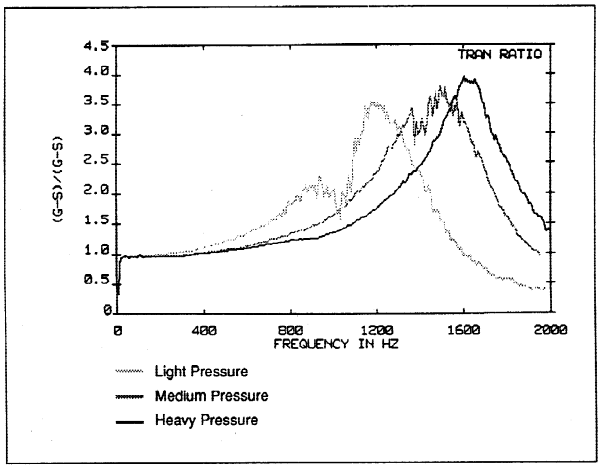


Figure 7. Comparisons, with respect to transfer function ratios, of the effects of hand pressure applied to a CSI 310 hand held probe with a 0.25 in. steel stinger attachment.

probe configuration. While reductions due to mounting looseness were not as severe as found in other mount configurations, the fact that tests were performed in a clean environment had some effect. Even though the stinger attachment was loose, the pressure applied to the probe eliminated some of the problems normally caused by slack joints and chatter. In field conditions, the technician experiences less than ideal conditions and vibrations occur in the vertical, horizontal and axial directions (not just vertically). A loose stinger connection in a field environment is likely to deteriorate the output signal more than it did in our laboratory tests because of the additional vibrations present in the horizontal and axial directions.

The hand pressure applied to hand held probes will vary between different people and among individuals. Therefore, tests were conducted to study the effect force variations have on hand held probes. For this experiment, two stingers, each connected to a CSI 310 hand held probe, were evaluated at three different degrees of force: light, medium and heavy (see Table 4 for results). Of the two stingers, the first stinger considered was a 10-32 UNF stud threaded tightly into the sensor until it extended 0.25 in. beyond the probe's tip. The frequency ranges for the light, medium and heavy pressure measurements were 3 to 395 Hz, 3 to 550 Hz and 3 to 605 Hz, respectively. For each decrement of hand force with respect to the heavily applied pressure measurement, the frequency ranges decreased 55 and 210 Hz. This changing value will vary with increases and decreases in hand pressure. Figure 7 compares the transfer function ratios for each of the three hand pressures.

When using the 8.5 in. aluminum stinger attachment, the acceptable frequency response ranged from 3 to 280, 3 to 320 and 3 to 415 Hz for the data collected using the hand held probe with light, medium and heavy hand forces, respectively. The frequency range decreased 95 and 135 Hz with decrease in vertical force.

The results in Table 4 show that the resonant frequency and therefore frequency range of the hand held probe varies proportionally with hand pressure. For measuring vibrations in the upper portion of the sensor's frequency range, consistent hand pressure is essential. Otherwise, erroneous readings may be obtained.

Data Collection Inconsistencies

Variability can occur due to measurement location, various people collecting data and individual inconsistency. The potential for introducing variability in the measured data was evaluated on bearing housings of rotating machinery utilizing typical data collection methods. A sketch showing the locations of measurement points for the two bearing housings studied appears in Figure 8. By collecting cross spectral data, all test data from each bearing housing could be compared and normalized with respect to its base data. The reference accelerometer used on the first bearing housing, which had a rounded surface, was a Wilcoxon 726T stud mounted (15 in-

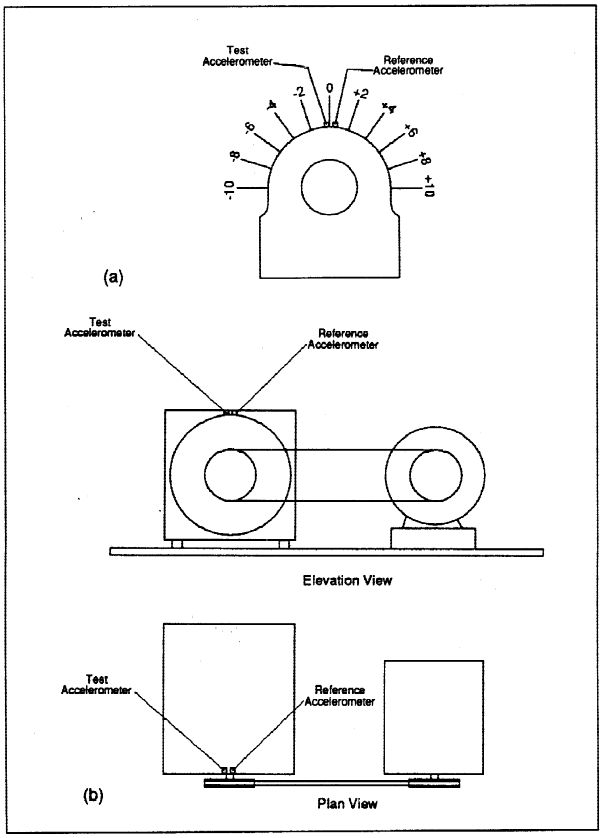


Figure 8. Schematic of: (a) bearing housing number 1 (numbers correspond to in. from the reference point 0); and (b) industrial machine driven by a DC motor (bearing housing number 2). Measurement points are indicated in each drawing.

lbs) to a CSI 906 super magnet and placed at position 0 as seen in Figure 8a. For measurements conducted on bearing housing number 2, which had a flat surface for mounting, the reference accelerometer (Wilcoxon 726T) was mounted with a rare earth magnet at the position indicated in Figure 8b.

Variability Due to Measurement Location. Data variability due to measurement point location was evaluated on bearing housing number 1 which contained a shaft turning at a speed of approximately 4 Hz. A Wilcoxon 726T accelerometer, also mounted on a CSI 906 super magnet, was used to collect the test measurements. A single spectrum was obtained at each

© Copyright 2001 Technical Associates Of Charlotte, P.C.

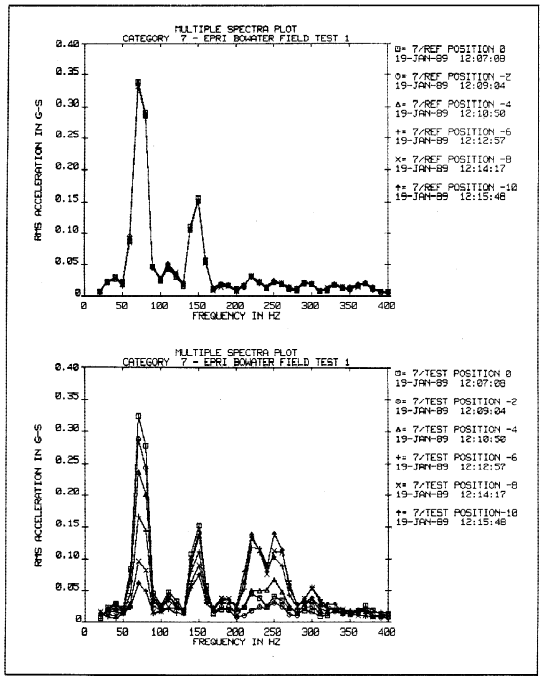


Figure 9. Spectra ranging from 0 to 400 Hz for reference (top graph) and test measurements (bottom graph) corresponding to data taken from the negative side of the bearing housing shown in Figure 8a.

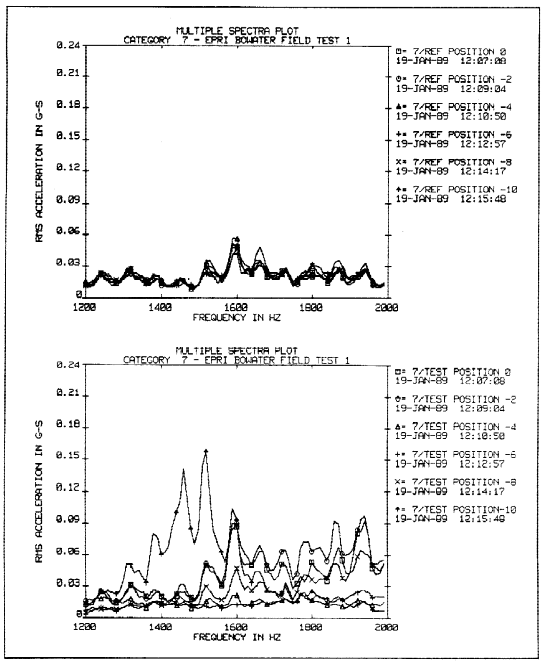
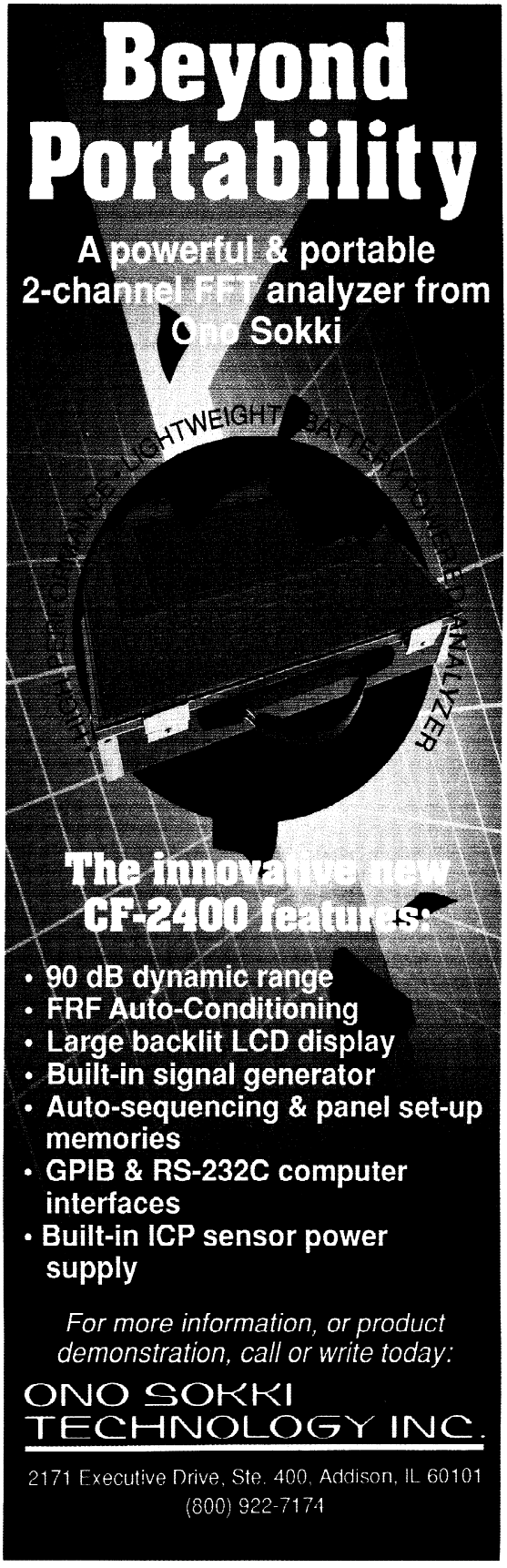


Figure 10. Spectra ranging from 1200 to 2000 Hz for reference (top graph) and test measurements (bottom graph) corresponding to data taken from the negative side of the bearing housing shown in Figure 8a.

measurement point over a frequency range of 0 to 2000 Hz by collecting twenty-five averages with 200 lines of resolution.

Comparisons of spectra computed from reference and test accelerometer measurements, taken from the negative side of bearing housing number 1, are displayed over the range of 0 to 400 Hz in Figure 9 and 1200 to 2000 Hz in Figure 10. Spectra (not shown) produced from the reference and test accelerometer readings were acquired on the positive side of the bearing



Circle 113 on Inquiry Card

© Copyright 2001 Technical Associates Of Charlotte, P.C.

housing yielding similar results. The frequency range between 400 and 1200 Hz is not displayed because data in this range were small in amplitude relative to the other frequencies.

Results showed that the magnitude for any given frequency was different for each change in measurement location. The largest magnitude at any given frequency occurred in the vertical and/or horizontal positions; i.e. some vibrations proved to be prominent vertically, while others were predominant horizontally. It was noted that a mere 2 in. displacement of the sensor on a housing approximately 40 in. in circumference would result in changed magnitudes ranging from 0.001 to 0.3 g depending on the frequency range evaluated. On other machines, the output change depends on the size of the vibrating surface. Larger surfaces may allow for more liberal movement of sensors while small machines may require a more specific measurement location in order to avoid collecting invalid data.

Measurement points should be chosen and marked in the vertical, horizontal and/or axial directions. The best marking method would be to mount (stud or adhesive) a washer or mounting disk onto the machine surface. Not only will these mounts be permanent, but the measuring surface will be flat. Another method that will improve the reliability of data is to mark the machine housing for each measurement location with a spot of paint or indention. In general, the more time and money put into marking measurement points, the more reliable the resulting data. One must be careful when choosing the position for measurement on a housing surface. Measurement points should be at positions having solid metal paths from the sensor to the source of vibration. Interfaces and internal voids can attenuate vibration and provide unreliable data.

Variability Resulting in Measurements by Different People. The variability in vibration readings taken by different people and the variability in repeated measurements by the same individual were evaluated on bearing housing number 2 which was running at a speed of 1440 rpm (24 Hz). All measurements were taken from the same marked point on the bearing housing in order to eliminate changes in measurement location as a factor contributing to the variability of the data. Base data were obtained using Wilcoxon 726T accelerometers. Both the reference and test accelerometer were stud mounted (torque of 15 in-lbs) on rare earth magnets.

All measurements taken from the bearing housing in Figure 8b were evaluated by calculating differences in magnitude of the reference accelerometer minus magnitude of the test accelerometer for prominent frequency ranges of concern. By comparing differences, the variance due to motor speed and reference sensor location was eliminated.

Variability introduced when different people collect data was assessed on the basis of the two tests described below:

1. Five individuals acquired acceleration measurements using a hand held probe (CSI 310) with a pointed 2 in. steel stinger attachment. Spectra with a frequency range of 0 to 2000 Hz were obtained with six averages and 400 lines of resolution.
2. Five individuals took acceleration measurements with the same Wilcoxon 726T accelerometer mounted on a rare earth magnet. Spectra with a frequency range of 0 to 2000 Hz were obtained with six averages and 400 lines of resolution.

In the first test, reference measurements were very similar and are not plotted. However, spectra calculated from test measurements are displayed in Figure 11. Energy within each spectrum was not evenly distributed over all frequencies, but was dispersed into groups within various frequency ranges. To compare the results of each person's measurements, the spectra were divided into major frequency ranges (or groups) which contained energy significant for evaluation. These groups are indicated by the vertical lines in Figure 11. Within each frequency range, an overall RMS acceleration value in g was obtained.

The largest spread in magnitude difference for any frequency group between 0 and 990 Hz was about 0.3 g. This value was for the range of frequencies between 885 and 990 Hz. Above 990 Hz, hand held probe measurements approached the resonant frequency of the sensor system. This phenomenon was characterized by large deviations between the reference accelerometer readings and individual readings.

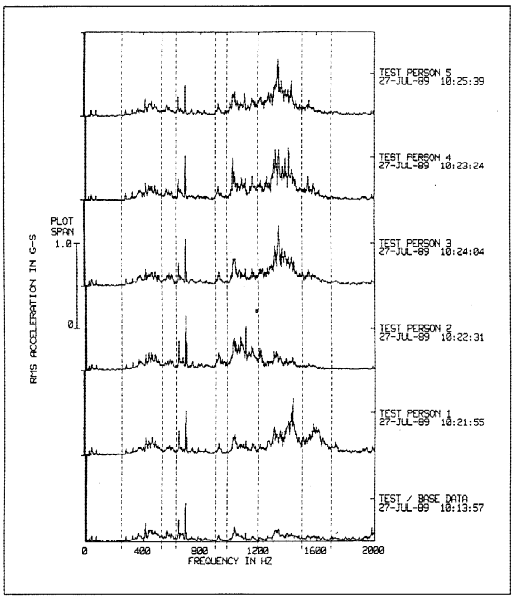


Figure 11. Test spectra comparing the measurements taken by five people using a CSI 310 hand held probe with a pointed 2 in. steel stinger attachment.

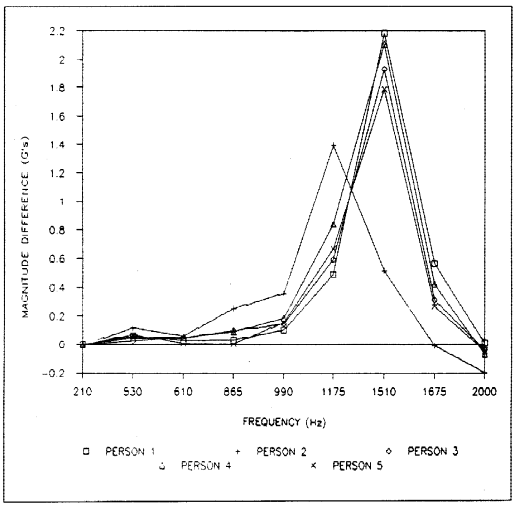


Figure 12. Deviation between the base and test magnitude difference curves derived from measurements taken with a CSI 310 hand held probe by five different people.

The previously reported results revealed the amount of variability (or non-repeatability) between readings by each individual. To determine the accuracy of the test sensor (hand held probe), the magnitude differences calculated for each measurement within a particular frequency group were subtracted from the magnitude differences calculated for the base data of the same frequency range. The base data were considered to be the "actual" or "true" vibration measurements. A graph showing the results of these calculations is seen in Figure 12.

The hand held probe resonant frequency changed with each individual's hand pressure. One measurement (person 2, Figure 12) was substantially different from the others. The resonant frequency for this reading was approximately 1175 Hz. Other resonant frequencies, for the same hand held probe, were in the neighborhood of 1510 Hz. These results demonstrate a problem that can be encountered when taking data

© Copyright 2001 Technical Associates Of Charlotte, P.C.

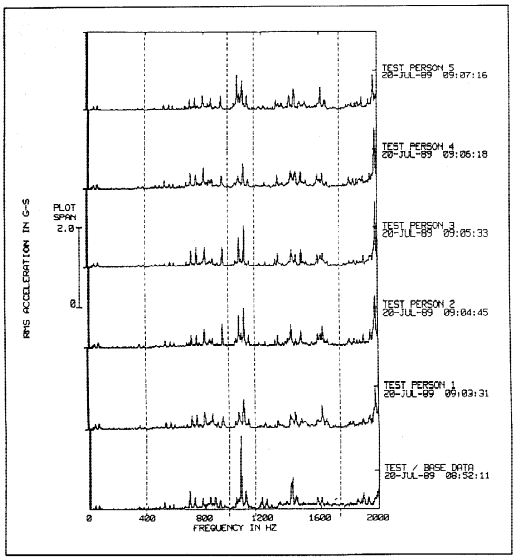


Figure 13. Test spectra comparing measurements between five people, each using the same Wilcoxon 726T accelerometer mounted on a rare earth magnet.

with a hand held probe, i.e. applied pressure to the probe will vary. The more it varies, the less repeatable and reliable the readings. Varying pressures applied to hand held probes are likely when different people take the same measurements. Even though the outlying point between 885 and 990 Hz (Figure 12) was present, the magnitude was only 0.4 g. Therefore, measurements taken by various individuals are accurate for frequencies up to 1000 Hz when using a hand held probe, as long as they are taken from the same measurement position.

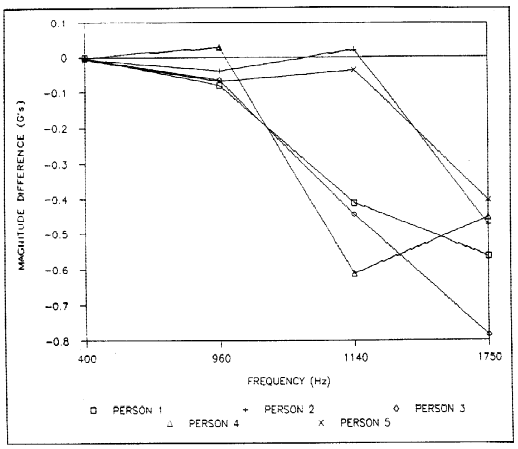
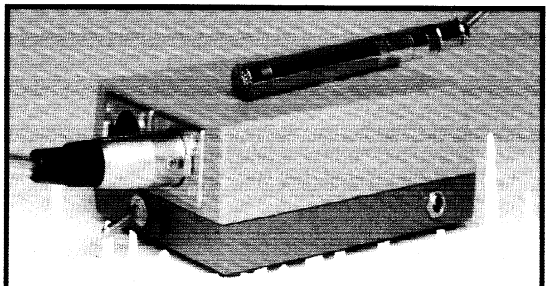


Figure 14. Deviation between the base and test magnitude difference curves derived from measurements taken by five different people with a Wilcoxon 726T accelerometer mounted on a rare earth magnet.

The second experiment included a test accelerometer (Wilcoxon 726T) mounted on a rare earth magnet. Spectra calculated from the test accelerometer measurements are plotted in Figure 13. Frequency ranges were evaluated as shown by the vertical boundary lines present in Figure 13.

Variations between the two major peaks around 1000 and 1100 Hz were observed in Figure 13. Smaller variations were also present at the frequencies ranging from 1300 to 1750 Hz.



INTERFACE

- INCLUDES selection of Type 1 measurement mikes, pre-amps and power supplies.
- COMPATIBLE with your choice of acoustic analysis equipment
- PROVEN RELIABILITY in architecture, noise control, sound synthesis, life sciences, product development, earth sciences, and many more applications
- LONG TERM STABILITY for temperature and humidity
- COST COMPETITIVE

ACOustics begins with

For full facts, call or write:

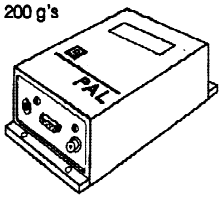
ACO Pacific, Inc.
2604 Read Avenue
Belmont, CA 94002 USA
(415) 595-8588 FAX: 591-2891

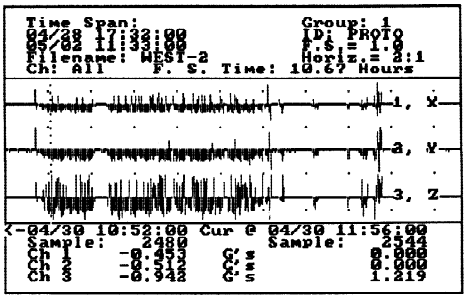
Circle 114 on Inquiry Card

PAL

Peak Acceleration Logger

Digital, 3 channels, IBM XT, AT programmable
8 ranges, 1, 2, 5, 10, 20, 40, 100, 200 g's
4 filters, 1 to 25, 50, 100, 200 Hz
Records 90 days on 6 "C" cells
Optional % RH, temp., LCD
10.3" x 5.9" x 3.5", 5.2 lbs.
Easy, fun, software included
Call for free demonstration disk





Dallas Instruments, Inc.
10205 Plano Road, Dallas, Texas 75238
(214) 349-1180 FAX (214) 348-7044

Toll free (800) 527-7071

Circle 115 on Inquiry Card

© Copyright 2001 Technical Associates Of Charlotte, P.C.

Since the reference and test accelerometers were mounted in the same configuration, most variations are due to differences in individual placement of the magnet on the marked spot, the offset location of the test accelerometer from the reference accelerometer and/or the inducement of rocking because of improper placement.

The largest magnitude difference for any individual measurement was 0.7 g which occurred between 1140 and 1750 Hz. With respect to accuracy, the frequency range between 1140 and 1750 Hz had the greatest deviation of 0.8 g (see Figure 14). This amount was not large enough to warrant limitation of the frequency range because 0.8 g is a relatively small change over the entire 610 Hz frequency range (1140-1750 Hz). Therefore, in "real world" applications, measurements taken by different people with a magnetically mounted accelerometer can be expected to vary 0.1 g for high energy frequency intervals less than 1000 Hz. Magnitude variation within high energy frequency segments may be as much as 0.6 g for measurements up to 2000 Hz.

Variability Resulting from Multiple Measurements by One Person. In industry, one person will often be responsible for collecting data. After hours of such measurements, fatigue can be transferred into carelessness which can then affect data reliability. Therefore, a test was administered to determine how repeatable and accurate a single person can measure vibrations over a period of time. Variability in repeated readings collected by the same individual was based on 20 measurements with a hand held probe (CSI 310) at a marked position on the bearing housing (see Figure 8b). Between each reading, the hand held probe was removed. For each measurement, a spectrum using six averages and 800 lines of resolution was acquired with a frequency range of 0 to 4000 Hz.

Spectra from the test measurements are plotted in Figure 15 for the base and first seven tests. The remaining 13 replications produce spectra with similar results and are not plotted.

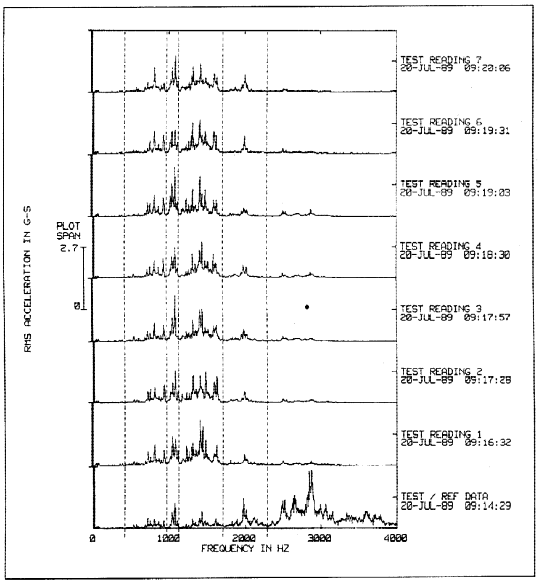
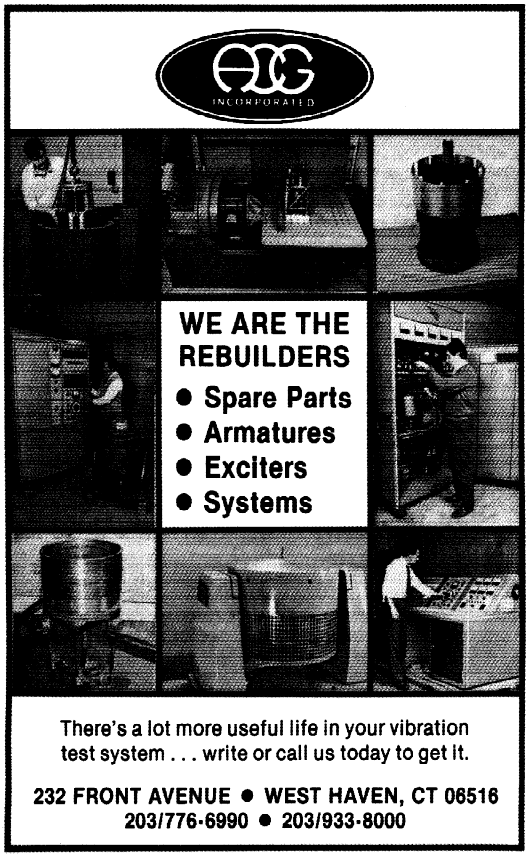


Figure 15. Test spectra of the first seven measurement replications taken by an individual using a CSI 310 hand held probe with a pointed 2 in. steel stinger attachment.

Data above 2300 Hz were not considered because very little energy was observed in this frequency region.

The largest spread in magnitude difference below 960 Hz was 0.3 g. As the resonant frequency of the hand held probe was approached, the measurements were not as accurate nor repeatable. However, when using a hand held probe (CSI 310) and measuring vibrations from a given point, data are repeat-



Circle 116 on Inquiry Card

Circle 117 on Inquiry Card

© Copyright 2001 Technical Associates Of Charlotte, P.C.

able and accurate from 0 to 960 Hz. Therefore, the results suggest that a person can collect valid data between 0 and 1000 Hz at scheduled intervals provided machines are properly marked for repetitive data acquisition. Furthermore, when collecting field data with a hand held probe, one can expect a measurement variation of 0.3 g for high energy frequency groups in the range of 0 to 1000 Hz. Employing more than one person to collect data can increase the variability.

Recommendations

Throughout this study, many mount configurations have been evaluated. Variations which can affect the response of sensors used in these configurations were investigated as well. The main theme derived from the results of these tests is to be consistent and methodical when acquiring data for spectral analysis. Below is a list of recommendations which will enhance the chances of obtaining accurate results. Recommendations are:

General

- ☐ Limit the measurement frequency range to the linear region of the mounted sensor system frequency response.
- ☐ Limit the number of different individuals recording data.
- ☐ Allow only one person to collect data for any given route.
- ☐ Always use the same sensor and mount configuration for a particular measurement point.
- ☐ Make measurements on clean, smooth and flat surfaces.
- ☐ Take measurements in horizontal, vertical and/or axial directions.
- ☐ Mark measurement points on machinery to eliminate variation due to measurement location.
- ☐ If possible, mount a steel disk on the machine at the measurement point.
- ☐ Tighten sensors (preferably to 15 in. lbs torque) to the mount configuration to avoid looseness. This rule applies to any configuration whether it includes a steel disk, a magnet, a quick lock or hand held probe.
- ☐ The more random the speed and load of a machine, the more averages required per measurement. For normal situations, 4 to 6 averages are sufficient. For vibrations that are more random, as many as 12 averages may be required.
- ☐ Avoid measurements where rocking may be induced. Adjust sensor position to eliminate rocking.

Magnet Mounting

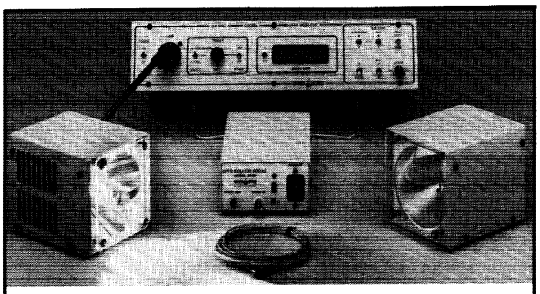
- ☐ Keep magnet bottom (feet) clean to avoid rocking and insufficient contact with the machine surface.
- ☐ Do not mount hand held probes on magnets because of possible cantilever action.
- ☐ Use as light a weight of cable as feasible. The heavier and stiffer the cable, the greater likelihood of cantilever action.

Hand Held Probe Use

- ☐ Keep a constant hand pressure for every measurement taken.
- ☐ The shorter the stinger attachment, for a particular vibrating structure, the better the frequency response.
- ☐ Do not allow a hand held probe to bounce while collecting data. Bouncing will be observed as chattering in the resulting spectra.
- ☐ Avoid tilting the probe. Take measurements so that the length of the probe is perpendicular to the surface of the machine being analyzed.

References

1. Serridge, M. and Licht, T. R., *Piezoelectric Accelerometer and Vibration Preamplifier Handbook*, Brüel & Kjær, Denmark, pp 28-29, 88-103, 1987.
2. Catlin, J. B., "A Survey of Factors Which Affect the Measured Vibration Spectra of Machines," IRD Mechanalysis, Inc., Columbus, OH.
3. Advanced Training Manual, IRD Mechanalysis, Columbus, OH, pp 61-66, 1980.



STOP THE ACTION SHOW THE MOTION...
WITH THE SLIP-SYNC® STROBEX®

Strobex produces a very bright, short duration flash of light which results in a sharply focused "stopped" or "slow motion" image of any subject. The Slip-Sync keeps the strobe image "locked on" while the subject rate changes and allows you to "stop" the subject at any position in the cycle. Ideal for analyzing the dynamics of a complex machine... or any subject from a bullet to a protozoa.

Typical applications

- Machine Vision
- Conventional Video Motion Analysis
- Slip-Sync Aided Video Analysis
- Sine Vibration Analysis
- Rotating Machinery Analysis
- Photo-Stress Analysis
- Machine Cutting Tool Analysis
- Cine Photomicroscopy

CHADWICK HELMUTH

4601 N. Arden Dr., El Monte, California 91731
(818) 575-6161, Telex 194-271, FAX 818-350-4236

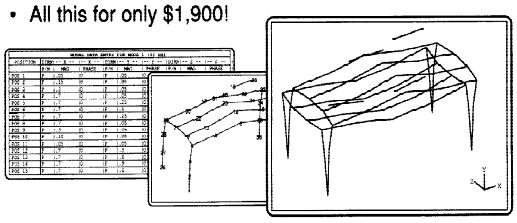
Innovations in industrial, scientific and medical strobe applications since 1954.

Circle 118 on Inquiry Card

AT LAST!
MODAL ANALYSIS
FOR THE
MAINTENANCE DEPARTMENT

ModalALERT is a no-fuss software tool designed to show you how a structure is vibrating. It helps your maintenance department solve many vibration and structural fatigue problems with an easy-to-use program on your 386 PC.

- Easy test setup
- Animation shows how the structure vibrates
- View rotation, scaling and other analysis aids
- Easy-to-understand reports on printer or plotter
- Use with most data collectors or spectrum analzers
- All this for only $1,900!



For a free demo diskette please contact:

DLI Engineering Corporation
253 Winslow Way West, Bainbridge Island, WA 98110
Phone: (206) 842-7656 Fax: (206) 842-7667

Circle 119 on Inquiry Card

© Copyright 2001 Technical Associates Of Charlotte, P.C.

References

1.John B. Catlin, A SURVEY OF FACTORS WHICH AFFECT THE MEASURED VIBRATION SPECTRA OF MACHINES, IRD MECHANALYSIS, INC.

2.S. V. Bowers, K. R. Piety, and R. W.Piety,REAL WORLD MOUNTING OF ACCELEROMETERS FOR MACHINERY MONITORING, Computational Systems, Incorporated, Knoxville, Tennessee, "Sound and Vibration Magazine" February 1991, Pages 14-23

© Copyright 2001 Technical Associates Of Charlotte, P.C.

CHAPTER 9

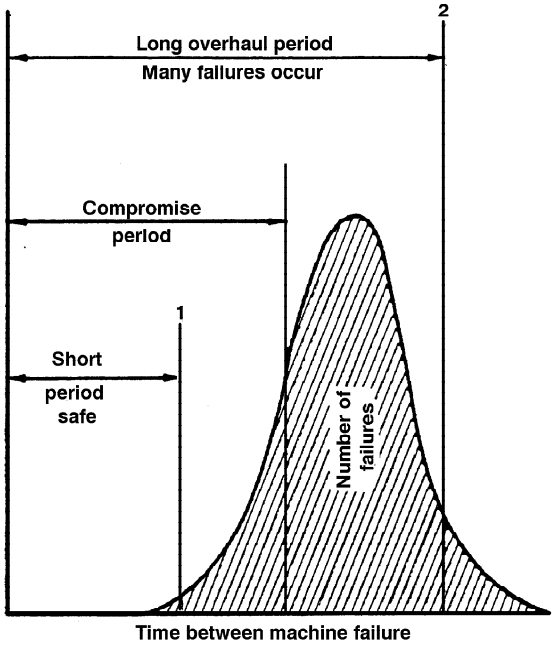
SETUP AND IMPLEMENTATION OF EFFECTIVE PREDICTIVE MAINTENANCE AND CONDITION MONITORING PROGRAMS

Condition Monitoring is concerned with extracting information from machines to indicate their condition, and to enable them to be operated and maintained with safety and economy.

BREAKDOWN MAINTENANCE: Machines can be run until they fail, and are then repaired. This obviously crude method of operation can be very expensive in terms of lost output and machine destruction, and it may also be dangerous.

REGULAR PREVENTIVE MAINTENANCE: A better method is to stop machines at regular intervals for maintenance in order to reduce the chance of unplanned stoppages, i.e. breakdowns. The choice of the best regular maintenance interval presents a difficult problem, with two opposite extremes to be avoided:

1) Too frequent maintenance wastes production time and increases the risk of trouble arising from human errors in reassembly:



REGULAR PREVENTIVE MAINTENANCE

on the other hand,

2) Too long an interval results in an unacceptable number of machine failures during operation.

A compromise between these two extremes can be established by experience, but machine failures will continue to occur.

CONDITION BASED MAINTENANCE: The best method is to carry out predictive maintenance at what may be irregular intervals but to determine these by the actual condition of the machine. Assessing the trend of vibration measurements can give a useful lead time in warning of incipient machine failure. Where considerable experience of a machine and its failure characteristics has been gained, a single check reading of its condition can provide sufficient data for determining a safe period of further running. Alternatively, an alarm indicator may be set to give warning when the safe running period has fallen to a critical level. Condition checking can also be used to compare the condition of similar machines. The readings taken may also be analyzed in detail to indicate the most likely cause of the problem.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

BENEFITS OBTAINABLE BY INDUSTRY: Benefits which can be derived from monitoring the condition of plant and machinery:

1) Increased plant availability resulting in greater output from the capital invested.
2) Reduces maintenance costs.
3) Improved operator and passenger safety.
4) More efficient plant operation and more consistent quality, obtained by matching the rate of output to the plant condition.
5) More effective negotiations with plant manufactures or repairers, backed up by systematic measurements of plant condition.
6) Better customer relations following from the avoidance of inconvenient breakdowns which would otherwise have occurred.
7) The opportunity to specify and design better plant in the future.

How these benefits arise:

1) Machine running time can be increased by maximizing the time between overhauls. Overhaul can be reduced because the nature of the problem is known, and the spares and men can be ready. Consequential damage can be reduced or eliminated.
2) The lead time given by condition monitoring enables machines to be stopped before they reach a critical condition, especially if instant shut-down is not permitted.
3) The operating load and speed on some machines can be varied to obtain a better compromise between output, and operating life to the next overhaul.
4) Measurements of plant when new, at the end of the guarantee period, and after overhaul, give useful comparative values.
5) The lead time given by condition monitoring enables such breakdowns to be avoided.
6) The recorded experience of the operation of the present machinery is used for this purpose.

ESTIMATING THE BENEFITS

THE BENEFITS: Vibration monitoring techniques can be easily applied to rotating (and reciprocating) machinery to detect most common problems, such as: unbalanced, bent or misaligned shafts; bearing or gear defects; belt, blade or impeller faults; structural, foundation or critical speed problems; rubbing, loss of lubrication and oil whirl.

The main savings of using predictive maintenance for these problems arise by avoiding losses of output due to the breakdown of machinery (about 65%) and by reducing the cost of maintenance (about 35%). Output related losses can be estimated from the number of days of lost production multiplied by the gross output value per day. The maintenance costs are rather more difficult to quantify, but related mainly to the labor costs of breakdown maintenance. As a very rough initial guide, industries are likely to achieve savings on the order of 0.5 to 3% of “added value output” (gross sales minus gross costs of raw materials).

Establishments which are particularly suitable will be those in which the output is produced by machinery rather than manual work, and which have a large output from one plant, so that potential breakdown losses are high. The level of savings is also likely to be at a maximum if the plant is currently incurring high costs for maintenance (greater than 80% of averaged annual capital investment). If maintenance is already well organized and breakdown costs still exceed 40% of total maintenance costs, predictive maintenance can also prove beneficial. In industries which involve an element of risk for employees (where 0.5% of more employees are killed or seriously injured) then predictive maintenance should be regarded as essential.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

ESTIMATING THE COSTS

THE COSTS: If an establishment carries out its own predictive maintenance, it will need to make an initial investment in equipment (40% of total) as well as training and experience acquisition (60% of total). There is also the ongoing costs of operating the program.

As a general guide for an industrial plant, a reasonable level of initial investment is 1% of total capital value of the plant monitored (5% for special safety considerations). The ongoing costs of operating a program depend on the approach but can be quite low if done on a small scale by existing personnel. The average cost is about 16% of gross savings. Establishments large enough to employ at least one person full time on the program will realize more success, because expertise at interpreting trends and readings can be maintained. Outside services for training personnel and initially setting up the program can prove financially wise, especially for small plants (fewer than 100 employees). This is because start-up costs are high and benefits long term. Upper management should be prepared to (1) put in the effort needed to organize and refine the predictive maintenance and (2) meet its costs for 1-2 years until the benefits become apparent.

FINAL ANALYSIS

SUMMARY: An accurate analysis of the costs and benefits of predictive maintenance should convincingly indicate its technical and economic importance to plant users and equipment manufacturers. The typical cost/benefit ratio of a well run program is **about $5.00 savings for every $1.00 expended.** Such profitability demands the interest of anyone concerned with modern industry and machinery.

REFERENCES AND CREDITS: This overview is based on:

1) A Survey of Condition Monitoring in British Industry and its Potential for wider Application. Michael Neale and Associates Report TRD 187 for the Department of Industry. June 1975.

2) An Updating of Report TRD 187 (1975) to Determine the Potential in 1978 for the Application of Condition Monitoring in British Industry. Michael Neale and Associates Report TRD 223 for the Department of Industry. April 1978.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

EXAMPLE DOCUMENTED SAVINGS FROM A SUCCESSFUL PREDICTIVE MAINTENANCE PROGRAM OPERATED BY TECHNICAL ASSOCIATES
(Report Excerpt)

TOTAL PMP PROGRAM COST SAVINGS DURING 1988-89:

Table I summarizes cost savings achieved during the 1988-89 contract in each of the major categories considered - maintenance labor and material savings; electrical energy cost savings; and production savings. It shows that during the first ten months, the total savings due only to maintenance costs and the electrical energy costs have been approximately $180,171 to date. This has been achieved with an investment of $44,580 for the full year to Technical Associates. Therefore, based on only the ten months to date, this represents a benefit/cost ratio of over 4.04; if the cost savings rate during the first ten months continues through the full year, Table I shows that these savings will total about $216,205, or a benefit/cost ratio of almost 4.85. If production savings are taken into account, this ratio should substantially increase to over 10 to 1.

TABLE I
ESTIMATED PMP PROGRAM COST SAVING
FOR 1987-88 and 1988-89 CONTRACTS

SAVINGS CATEGORY	1987-88	1988-89 SAVINGS	
		FIRST 10 MONTHS	ANNUALIZED RATE
A. Maintenance Labor & Material Savings	$ 86,496	$ 134,385	$ 161,262
B. Electrical Energy Cost Savings	$ 28,389	$ 45,786	$ 54,943
SUBTOTAL	**$ 114,885**	**$ 180,171**	**$ 216,205**
C. Production Saving (Increased Availability)	$ 172,992	$ 268,770	$ 322,524
TOTAL ESTIMATED CONTRACT SAVINGS	**$ 287,877**	**$ 448,941**	**$ 538,729**
D. Investment with Technical Associates	$ 49,368	$ 37,150	$ 44,580

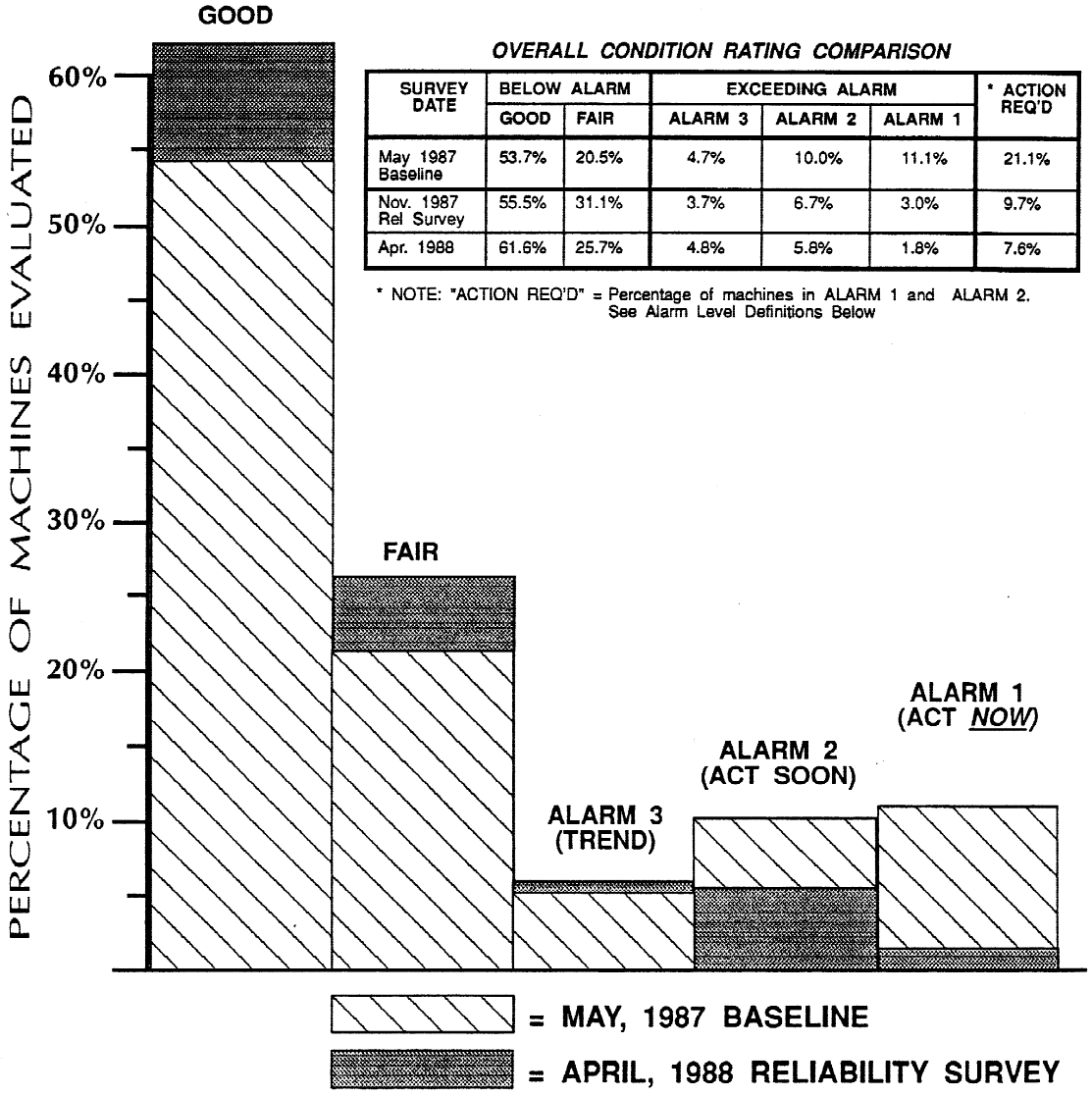
CLOSING

We at Technical Associates are very pleased with the results to date on our 1988-89 Predictive Maintenance contract. We feel strongly this has been due to the excellent joint cooperation between our two companies including the continuing action you have taken during the year when monthly recommendations were provided by Technical Associates. Frankly, we consider this program at your company our **"Model Predictive Maintenance Program"** and would hope that all our clients would achieve this magnitude of savings by the constant reaction of their maintenance departments to recommendations offered in the **PMP** reports.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

FIGURE 1

COMPARISON OF OVERALL CONDITION RATING SINCE IMPLEMENTATION OF PLANT PREDICTIVE MAINTENANCE PROGRAM (For Machines in Areas 5, 6, and 7)



OVERALL CONDITION RATING COMPARISON

SURVEY DATE	BELOW ALARM		EXCEEDING ALARM			* ACTION REQ'D
	GOOD	FAIR	ALARM 3	ALARM 2	ALARM 1	
May 1987 Baseline	53.7%	20.5%	4.7%	10.0%	11.1%	21.1%
Nov. 1987 Rel Survey	55.5%	31.1%	3.7%	6.7%	3.0%	9.7%
Apr. 1988	61.6%	25.7%	4.8%	5.8%	1.8%	7.6%

* NOTE: "ACTION REQ'D" = Percentage of machines in ALARM 1 and ALARM 2. See Alarm Level Definitions Below

ALARM DEFINITIONS:

- ***ALARM 1*** = MAINTENANCE ACTION RECOMMENDED ASAP. (SEE "RECOMENDATIONS/REMARKS" FOR THIS MACHINE)
- ***ALARM 2*** = SCHEDULE REPAIRS AT CONVENIENCE. (SEE "RECOMENDATIONS/REMARKS" FOR THIS MACHINE)
- ***ALARM 3*** = NO MAINTENANCE ACTION REQUIRED AT THIS TIME (CONTINUE ONLY WITH RELIABILITY SURVEY TREND ANALYSIS)

© Copyright 2001 Technical Associates Of Charlotte, P.C.

ATTACHMENT II.
POTENTIAL ANNUAL ELECTRICAL ENERGY SAVINGS FROM PREDICTIVE MAINTENANCE PROGRAM

ASSUMPTIONS:

1. Assuming 1.6% Reduction in Motor Current Driving Load due to decreased vibration-induced dynamic loading (from unbalance, misalignment, worn components and other problems which were detected and corrected). Current National Average shows approximately 1.6% to 2.4% energy reduction for those plants which fully implement Corrective Condition Monitoring Programs.

2. Assuming 4.6 c/kWHr Electricity Cost

3. Assuming 171 electrical motors with average capacity of 66.83 HP for those machines now in program (from actual survey of HP of each motor).

4. Assuming 24 hr/day operation year-round .

A. ANNUAL ENERGY COSTS REQUIRED FOR 1 MOTOR OF 66.83 HP RUNNING 24 HR/ DAY (8760 HRS/YR) AT $.046/kWH:

$$= (66.83\ HP)\ (.7457\ \frac{kw}{HP})\ (8760\ \frac{hrs}{yr})\ (\frac{\$.046}{kWHr}) = \frac{\$20,082}{\text{per motor}}$$

B. ANNUAL ENERGY COSTS FOR 171 MOTORS OF 66.83 HP AVERAGE SIZE:

$$= (\frac{\$20,082}{\text{yr-motor}})\ (171\ \text{motors}) = \frac{\$3,434,022}{yr}\ (\text{for 171 motors})$$

(74,651,033 kWHr/yr)

C. PMP ENERGY SAVINGS FOR A 1.6% MOTOR CURRENT REDUCTION:

$$= (\frac{\$3,434,022}{yr})\ (.016) = \frac{\$54,943/yr}{}$$ (Estimated Annual Electrical Energy Savings on 171 PMP Motors)

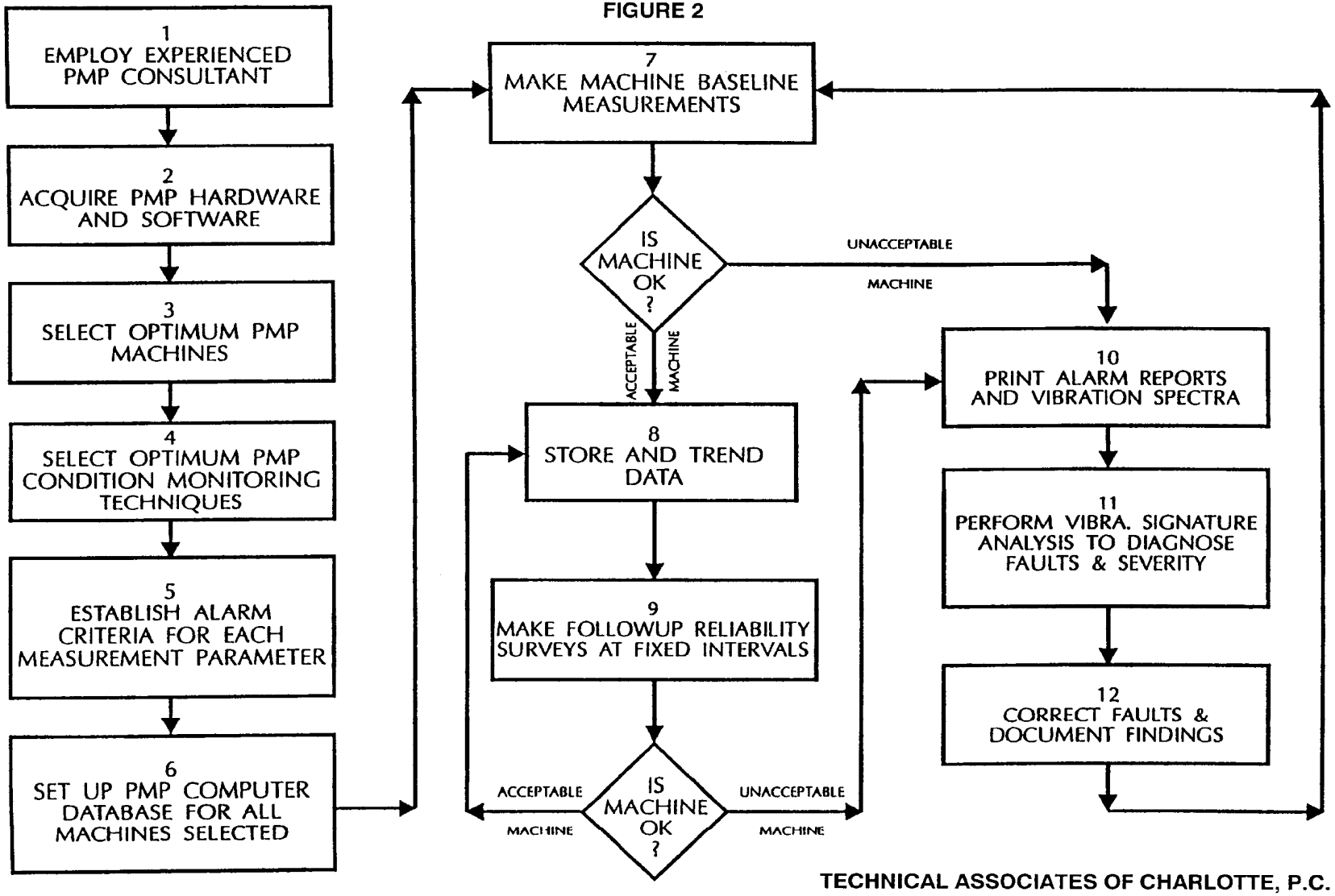
($45,786 during first 10 months of year)

© Copyright 2001 Technical Associates Of Charlotte, P.C.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

RECOMMENDED PREDICTIVE MAINTENANCE PROGRAM FLOW CHART

FIGURE 2



TECHNICAL ASSOCIATES OF CHARLOTTE, P.C.

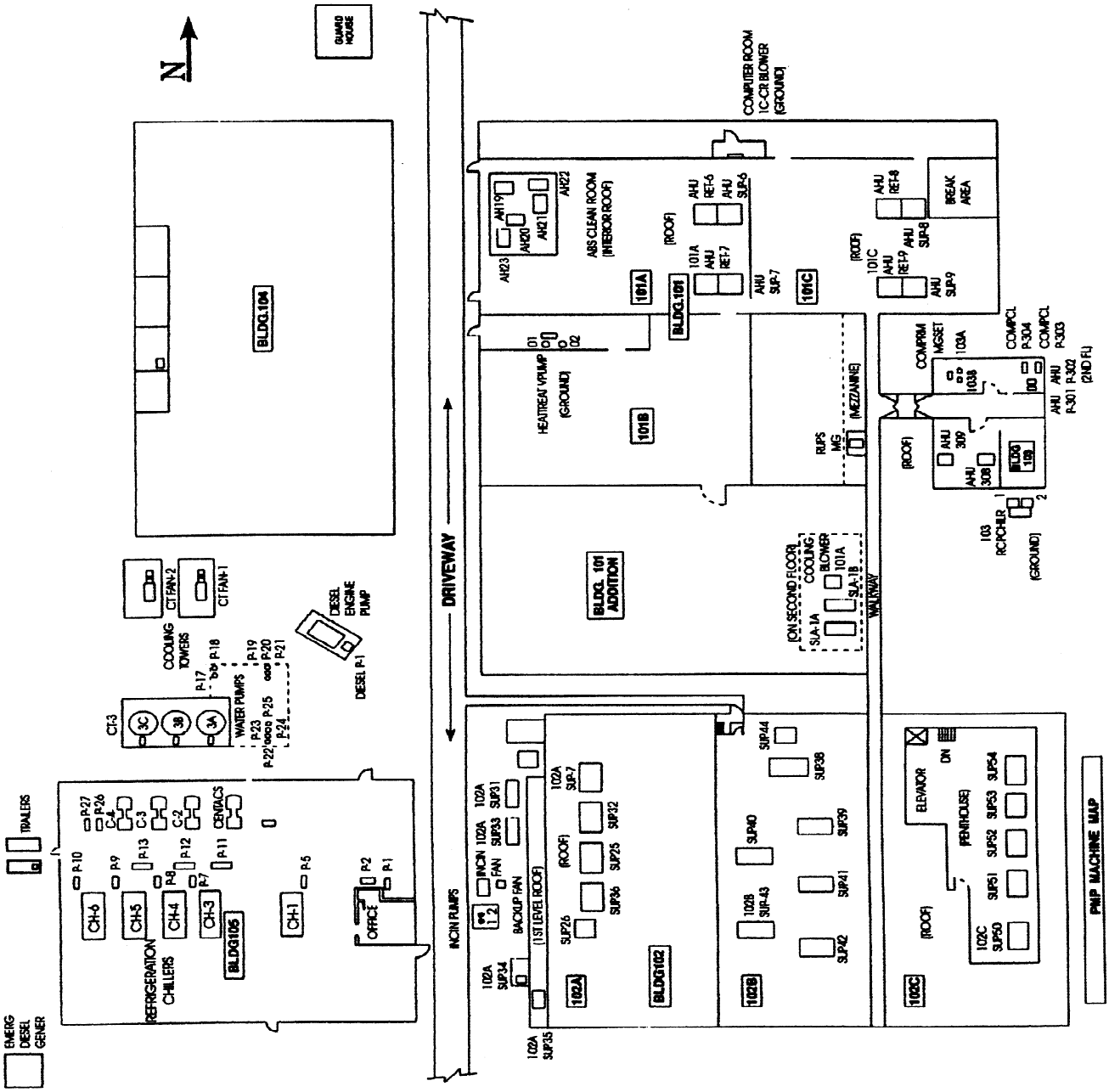


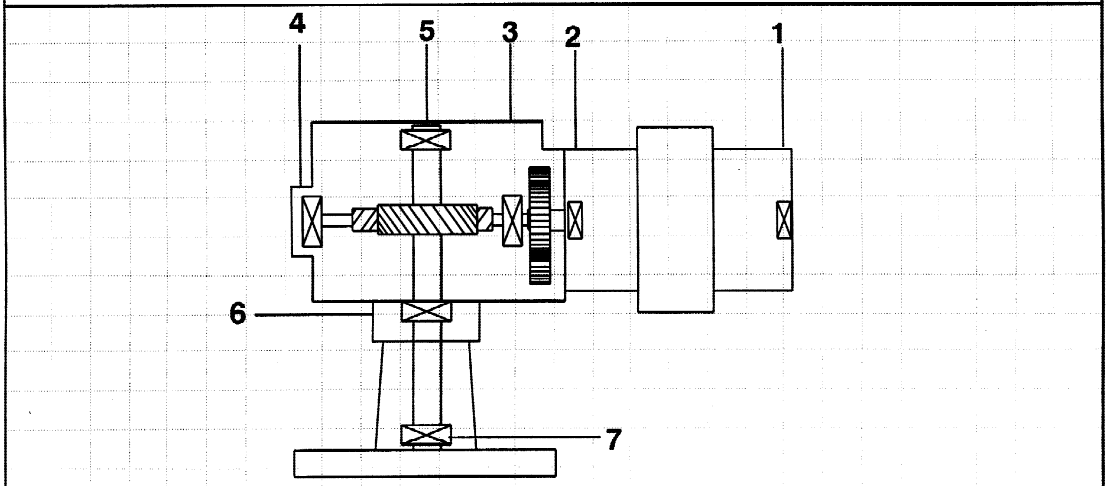
FIGURE 3
SAMPLE PMP MACHINE PLANT LAYOUT

© Copyright 2001 Technical Associates Of Charlotte, P.C.

Technical Associates of Charlotte, P.C.
(704) 333-9011

MACHINERY DATA GUIDE SHEET

RELIABILITY SURVEY predictive maintenance program (PAGE___OF___)



MACHINE NAME: #1 AGITATOR PRO1 **BLDG.:** 124
COMPUTER NAME: _____ **FLM #:** 269PA06000

DRIVER	INTERMEDIATE	DRIVEN
RPM min: **1760**	RPM min:	RPM min:
max:	max:	max:
MFGR:	RATIO: **(1.8:1 10:1) 18:1**	MFGR: **PFAUDLER**
MODEL:	MODEL: **FMDWV50600-EJD (5RW)**	MODEL:
SER NO.:	SER NO.: **88099165-002**	SER NO.:
___Volts; ___Amps; ___Phase	RATING:	RATING:
SHEAVE PITCH DIA.:	BELT LENGTH:	#BLADES OR VANES:
*BRG#: (IB)	*BRG#: (IB)	*BRG#: (IB)
*BRG#: (OB)	*BRG#: (OB)	*BRG#: (OB)
HP: **25**	#INPUT GEAR TEETH:	COUPLING TYPE:
#ROTOR BARS:	#OUTPUT GEAR TEETH:	OTHER:

***NOTE:** If Bearing is a Plain (or Sleeve) Bearing, simply enter "Sleeve" where shown.

R-112492-1

FIGURE 4

© Copyright 2001 Technical Associates Of Charlotte, P.C.

TECHNICAL ASSOCIATES OF CHARLOTTE, P.C.

CRITERIA FOR OVERALL CONDITION RATING (PEAK OVERALL VELOCITY,mm/sec)*

1. Assuming Machine Speed = 600 to 60,000 RPM.
2. Assuming Measurements by Accelerometer or Velocity Pickup securely mounted as Close as Possibleto Bearing Housing.
3. Assuming Machine Is Not Mounted on Vibration Isolators (for Isolated Machinery - Set Alarm 30% - 50% Higher).
4. Set Motor Alarms the Same as that for the Particular Machine Type unless Otherwise Noted.
5. Consider Setting Alarms on Individual External Gearbox Position about 25% Higher than that for a particular Machine Type.

			(WARNING)	(FAULT)
MACHINE TYPE	**GOOD**	**FAIR**	**ALARM 1**	**ALARM 2**
COOLING TOWER DRIVES				
Long, Hollow Drive Shaft	0 - 7.0	7.0 - 11.0	11.0	16.0
Close Coupled Belt Drive	0 - 5.0	5.0 - 7.5	7.5	12.0
Close Coupled Direct Drive	0 - 3.5	3.5 - 5.5	5.5	8.0
COMPRESSORS				
Reciprocating	0 - 6.0	6.0 - 9.0	9.0	13.5
Rotary Screw	0 - 5.5	5.5 - 8.0	8.0	12.0
Centrifugal With or W/O External Gearbox	0 - 3.5	3.5 - 5.5	5.5	8.0
Centrifugal - Integral Gear (Axial Meas.)	0 - 3.5	3.5 - 5.5	5.5	8.0
Centrifugal - Integral Gear (Radial Meas.)	0 - 2.5	2.5 - 4.5	4.5	6.5
BLOWERS (FANS)				
Lobe-Type Rotary	0 - 5.5	5.5 - 8.0	8.0	12.0
Belt-Driven Blowers	0 - 5.0	5.0 - 7.5	7.5	11.5
General Direct Drive Fans (with Coupling)	0 - 4.5	4.5 - 6.5	6.5	10.0
Primary Air Fans	0 - 4.5	4.5 - 6.5	6.5	10.0
Vacuum Blowers	0 - 3.5	3.5 - 5.5	5.5	8.0
Large Forced Draft Fans	0 - 3.5	3.5 - 5.5	5.5	8.0
Large Induced Draft Fans	0 - 3.0	3.0 - 5.0	5.0	7.0
Shaft-Mounted Integral Fan (Extended Motor Shaft)	0 - 3.0	3.0 - 5.0	5.0	7.0
Vane-Axial Fans	0 - 2.5	2.5 - 4.5	4.5	6.5
MOTOR/GENERATOR SETS				
Belt-Driven	0 - 5.0	5.0 - 7.5	7.5	12.0
Direct Coupled	0 - 3.5	3.5 - 5.5	5.5	8.0
CHILLERS Reciprocating	0 - 4.5	4.5 - 7.0	7.0	10.5
Centrifugal (Open-Air) - Motor & Compressor Separate	0 - 3.5	3.5 - 5.5	5.5	8.0
Centrifugal (Hermetic) - Motor & Impellers Inside	0 - 2.5	2.5 - 4.0	4.0	6.0
LARGE TURBINE/GENERATORS				
3600 RPM Turbine/Generators	0 - 3.0	3.0 - 5.0	5.0	7.0
1800 RPM Turbine/Generators	0 - 2.5	2.5 - 4.0	4.0	6.0
CENTRIFUGAL PUMPS				
Vertical Pumps (12' - 20' Height) } Height from grade to Top Motor Bearing.	0 - 6.0	6.0 - 9.0	9.0	13.5
Vertical Pumps (8' - 12' Height) } May be Necessary To Spec. Lower Alarm	0 - 5.0	5.0 - 7.5	7.5	11.5
Vertical Pumps (5' - 8' Height) } for Lower Motor Bearings & the Upper	0 - 4.0	4.0 - 6.0	6.0	9.5
Vertical Pumps (0' - 5' Height) } Pump Bearing (depending on height).	0 - 3.5	3.5 - 5.5	5.5	8.0
General Purpose Horizontal Pump - Direct Coupled	0 - 3.5	3.5 - 5.5	5.5	8.0
Boiler Feed Pumps - Horizontal Orientation	0 - 3.5	3.5 - 5.5	5.5	8.0
Piston Type Hydraulic Pumps - Horizontal Orientation (under load)	0 - 2.5	2.5 - 4.5	4.5	6.5
MACHINE TOOLS				
Motor	0 - 1.8	1.8 - 3.0	3.0	4.5
Gearbox Input	0 - 2.5	2.5 - 4.0	4.0	6.0
Gearbox Output	0 - 1.6	1.6 - 2.5	2.5	4.0
Spindles:				
a. Roughing Operations	0 - 1.2	1.2 - 1.8	1.8	2.5
b. Machine Finishing	0 - .72	.72 - 1.1	1.1	1.6
c. Critical Finishing	0 - .45	.45 - .72	.72	1.1

***NOTE: The "ALARM 1" and "ALARM 2" overall levels given above apply only to in-service machinery which has been operating for some time after initial installation and/or overhaul. They do not apply (and are not meant to serve as) Acceptance Criteria for either new or rebuilt machinery.**

COPYRIGHT© 2000 - TECHNICAL ASSOCIATES OF CHARLOTTE, P.C.

R-200002-RMSMM

© Copyright 2001 Technical Associates Of Charlotte, P.C.

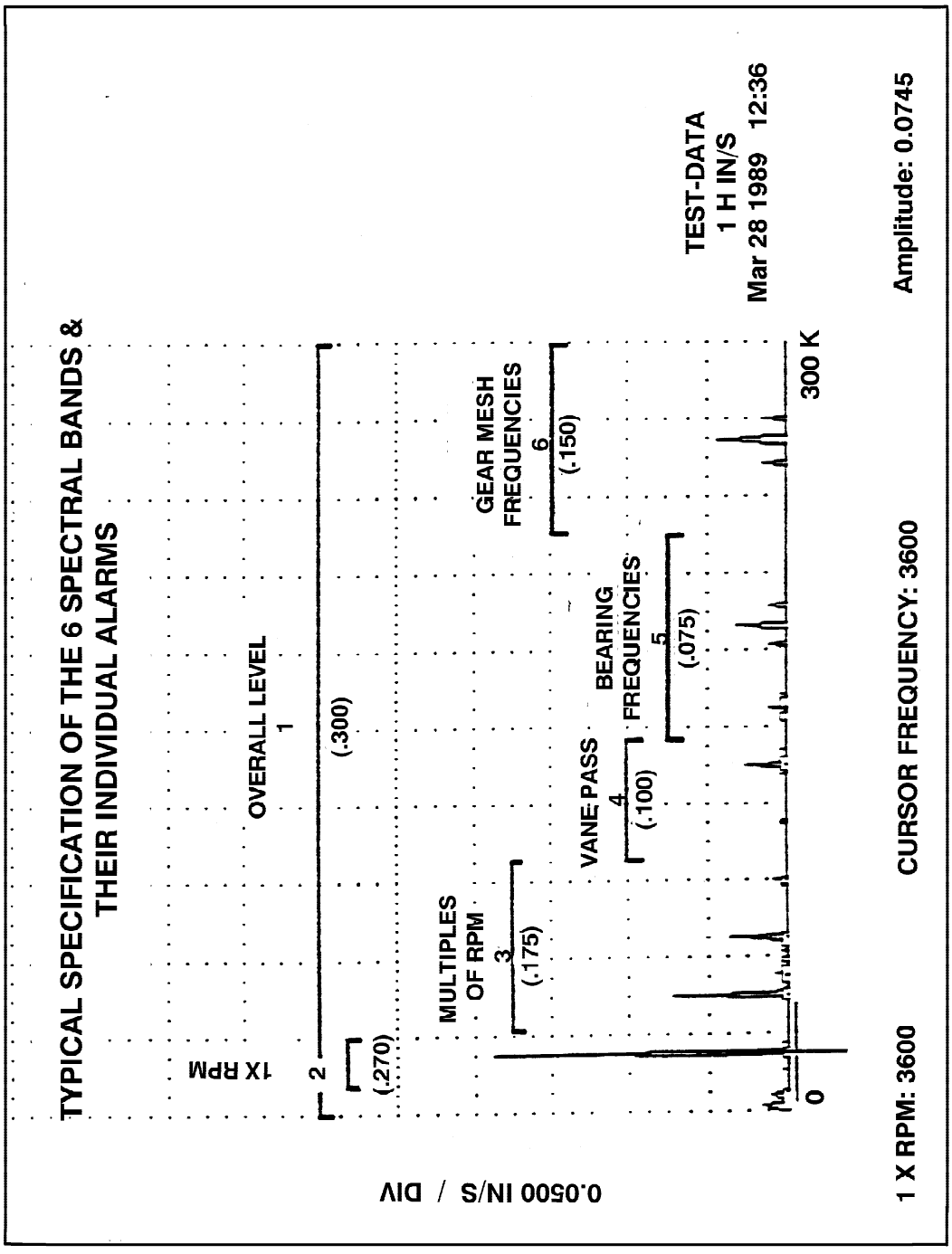


FIGURE 5

© Copyright 2001 Technical Associates Of Charlotte, P.C.

FIGURE 6 DESCRIPTION
OVERALL CONDITION RATING SUMMARY CHART

An overall condition rating of each machine covered under the **PREDICTIVE MAINTENANCE PROGRAM** is provided for quick reference. This condition rating is determined by rigorous review of **Reliability Survey** vibration data including trends of overall levels, trends of selected key frequencies, analysis of vibration spectra and review of " waterfall spectra" (showing how vibration signatures change from one survey to the next).

Next, an assessment is made of the severity of any problems detected during the latest **Reliability Survey**. Both the problem found and the recommended solutions are included in the **"Rank-Ordered Summary of Results and Recommendations for Problem Machines"** chart. On this chart, the severity of each problem and when it should be resolved is listed in order of highest to lowest problem severity.

Each of these levels of recommended action are then repeated in this "Overall Condition Rating Summary Chart" with headings of "Trend Problem Only"; "Schedule Repair at Convenience" ; and "Maintenance Required ASAP". Nothing needs to be done by plant maintenance at this time for those machines listed under "Trend Problem Only". Simply refer to the problem detected and watch for any deterioration of this problem in future **Reliability Surveys**.

It is strongly recommended that maintenance action be taken at the earliest possible moment for those machines listed under the column "Maintenance Required ASAP". These machines have been found to have serious problems which might cause catastrophic failure in the near future. Recommended actions are listed in the aforementioned results and recommendations chart.

Maintenance should be scheduled when possible for those machines listed under "Schedule Repair at Convenience". In general, these machines have less serious problems at the time of the **Reliability Survey**, but still mandate close attention for any possible deterioration. Again, recommended actions are listed in the results and recommendations summary chart.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

AS OF DECEMBER 8, 1992

	MACHINES NOT IN ALARM		MACHINES WITH PROBLEMS DETECTED (IN ALARM)		
MACHINE LOCATION (TRAIN)	GOOD	FAIR	Trend Problem Only (Priority 3)	Schedule Repair at Convenience (Priority 2)	Maintenance Required ASAP (Priority 1)
BLD-101A	101A ABS AH-21 101A RETN AHU-6 101A RETN AHU-7	101A ABS AH-19	101A ABS AH-22 101A SUPY AHU-6 101A SUPY AHU-7 SLA BLOWER 1A	101A ABS AH-20 1C-CR BLOWER	101A ABS AH-23
BLD-101B			HEATREAT VPMP-1		
BLD-101C			101C RETN AHU-8 101C SUPY AHU-8 101C SUPY AHU-9	101C RETN AHU-9	
BLD-102A		102A SUPY AH-31 102A SUPY AH-7	102A SUPY AH-25 102A SUPY AH-26 102A SUPY AH-32 102A SUPY AH-33 102A SUPY AH-35	102A SUPY AH-34 102A SUPY AH-36	
BLD-102B		102B SUPY AH-41 102B SUPY AH-43	102B SUPY AH-38 102B SUPY AH-40 102B SUPY AH-42		102B SUPY AH-44
BLD-102C		102C SUPY AH-51 102C SUPY AH-52	102C SUPY AH-50 102C SUPY AH-53 102C SUPY AH-54		
BLD-103	103 AHUSYS P301 103 AHUSYS P302 103 COMPCL P303	CAFETER AHU-309	103 COMPCL P304 103 RCP CHILR 1 103 RCP CHILR 2 103B COMPRM GEN	CAFETER AHU-308	
BLD-105	#3 CHILR SUP P7 #3 YORK CHILLER #6 CHIL SUP P10 CENTAC COMP 4AF CH WTR CIRC P12 COMP COOLING P26 COMP COOLING P27 COOL TWR VPMP17 COOL TWR VPMP18 COOL TWR VPMP19 COOLING TOWER 02 VS CWTR CRC P11 VS HOTWATER P-1	CENTAC COMP-2 CENTAC COMP-3	#6 YORK CHILLER CENTAC COMP-1AF CH WTR CIRC P13 COND WTR VPMP22 COND WTR VPMP23 COND WTR VPMP24 COOL TWR VPMP20 COOL TWR VPMP21 COOLING TOWER 3A COOLING TOWER 3C DIESEL ENG PUMP DIESEL P-1 EMERG DIESL GEN HOT WATER P-28	COOLING TOWER 01 COOLING TOWER 3B	COND WTR VPMP25
INCIN	INCIN VA FAN		INCIN HYD PMP 1		

For Various Reasons, The Following Machines Were Not Operating or Measured

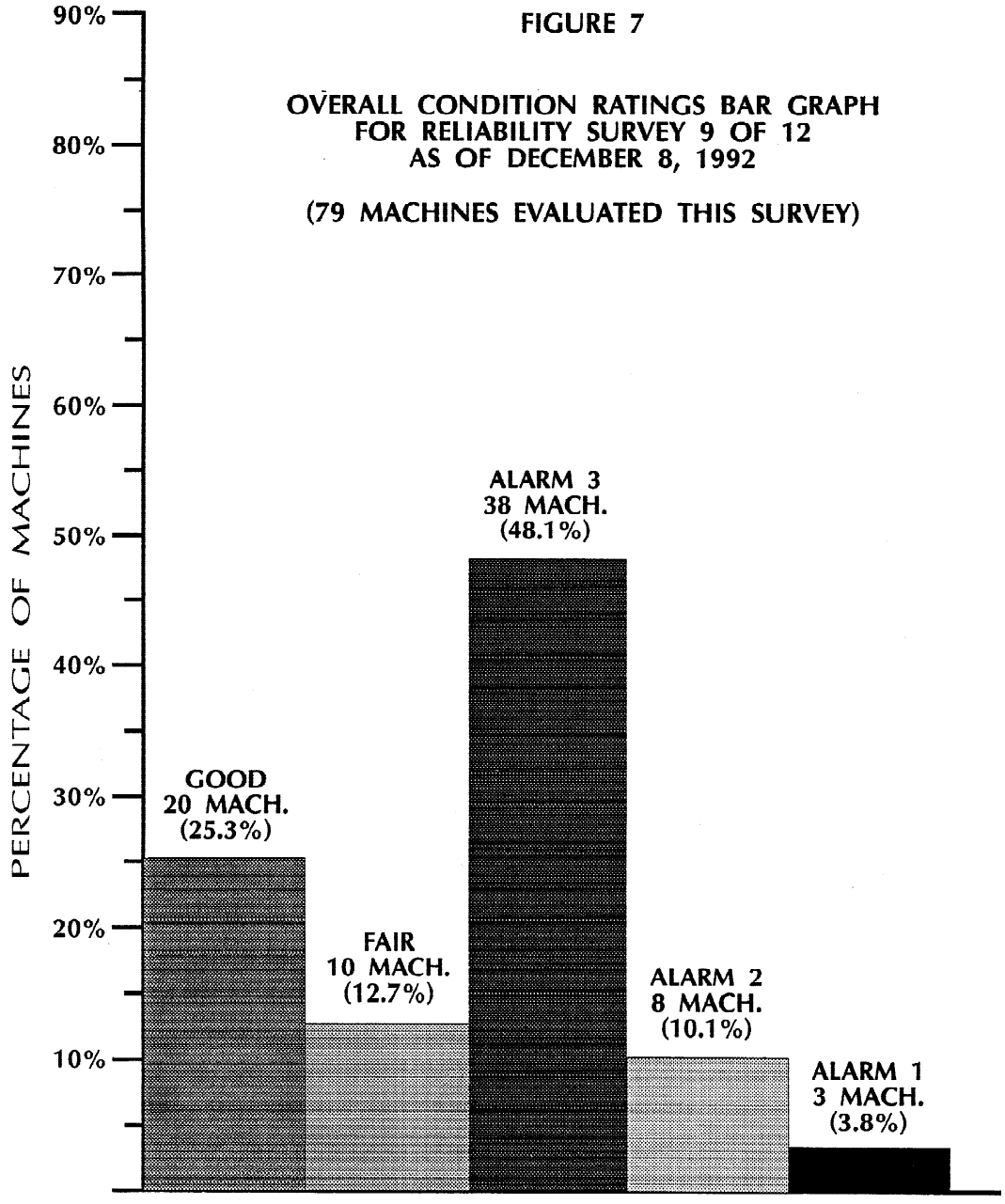
#3 CHILR SUP P5	103A COMPRM GEN
#1 VS YORK CHIL	COOL BLWER 101A
#4 CHILR SUP P8	HEATREAT VPMP2
#4 YOR CHILLER	HOT WTR SUP P-2
#5 CHILR SUP P9	INCIN BCKUP FAN
#5 YORK CHILLER	INCIN HYD PMP 2

FIGURE 6
OVERALL CONDITION RATING REPORT

© Copyright 2001 Technical Associates Of Charlotte, P.C.

FIGURE 7 DESCRIPTION
OVERALL CONDITION RATINGS BAR GRAPH

This chart provides a quick glance for PMP personnel and managers into the current state of the plant's machinery condition without getting bogged down in the details of the machine problem or which machines are in alarm. It is useful to trend this information from survey to survey and note if the number of GOOD and FAIR machines grows while the number of machines in ALARM reduces. This provides a useful tool for PMP justification or for future program enhancements.



© Copyright 2001 Technical Associates Of Charlotte, P.C.

FIGURE 8 DESCRIPTION
RANK ORDERED SUMMARY OF RESULTS AND RECOMMENDATIONS FOR PROBLEM MACHINES

Problems detected during the **Reliability Survey** are summarized in this section along with Recommendations required to rectify them. Importantly, these are **RANK ORDERED** beginning with the machines considered to have the most severe problems, down to the machines having less serious problems. Not only is each problem identified, but also a priority number (from 1 to 4 with 1 being of highest severity) is used to assess the problem severity and when action should be taken, if any. Also listed is the highest overall vibration position and the measurement at that position for each machine.

Remarks and Recommendations are provided on the right-hand-side explaining what is required to successfully repair this machine. The last column lists the plot number(s) for spectra supporting the problem diagnosis.

It is of great importance that the plant follow these recommendations in order that it might achieve fullest possible returns from its **PREDICTIVE MAINTENANCE PROGRAM**. In general, it is recommended that the machines be repaired in the order listed beginning with the #1 problem priority machines. These machines have such high levels of vibrations that they may suffer catastrophic failure if action is not taken quickly. #2 problem priority machines should next be acted upon. Although these machines also have rather significant vibration levels or noticeable component wear problems, it is felt that the Maintenance Department can simply schedule these at their convenience. #3 problem machines require only minor inspection/corrective/diagnostic action such as inspect for loose or broken fasteners, lubricate bearing, or obtain phase data. Note that no action is required by Maintenance on #4 problem priority machines where problems detected on these machines require only trending in future **Reliability Surveys** looking for possible condition deterioration.

In general, it is recommended that neither scheduled nor unscheduled repairs be performed on machinery indicated to be in good operating condition by the PMP program. This will tend to waste expensive maintenance dollars and reduce efficiency both of the program and of the Maintenance staff. However, normal repairs such as periodic lubrication, filter changes, cleaning, fastener tightening, etc. should be continued.

When repairs are made on machines, it is of great importance that the plant notify Technical Associates when the repair was made and what action was taken. This will allow one to evaluate the effectiveness of the repair and will determine whether all problems have been rectified.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

SUMMARY OF RESULTS AND RECOMMENDATIONS FOR MACHINES FOUND TO HAVE PROBLEMS

(SORTED FIRST BY PROBLEM PRIORITY; THEN BY PROBLEM SEVERITY WITHIN PRIORITY) AS OF:

MACHINE COMPUTER NAME	INDICATED PROBLEM(S): UNBALANCE	MISALIGNMENT/SOFT FOOT	BENT SHAFT	LUBRICATION PROBLEM	BEARING PROBLEM	MECHAN. LOOSENESS	BELT/SHEAVE PROBLEM	RESONANCE PROBLEM	GEAR PROBLEM	BLADE PASSING PROB.	CAVITATION	MOTOR ELECTRICAL	OIL WHIRL /WHIP	LOOSE FOUNDATION	VIB. MONITOR PROB.	SEAL /ROTOR RUB	OTHER (SEE REMARKS)	*PRIORITY OF PROBLEM	HIGHEST OVERALL VIBRATION: POSITION NO.	LEVEL (m = mils, l = in/sec, g = g's)	RECOMMENDATIONS & REMARKS	SEE PLOT NO.
102B SUPY AH-44					●	●	●	●						●				①	1A 2A 3BF	1.74i 1.51i .740i	The repair log states vibration pads and bracing for the blower were installed. However, high fan speed continues to be measured on the motor due to fan sheave irregularities and likely a weak or resonant motor mounting system. Both motor bearings have indications of rolling element bearing race defects. Inboard fan bearing has peaks which may be belt frequency due to loose belts or the inboard fan bearing turning in its housing or loose on the shaft. Recommend inspecting both fan bearings for proper fits and wear. The motor foundation and fan inboard bearing support obviously need structural stiffening due to weak or resonant structure or loose fasteners. Verify sheaves do not have any excess run-out and replace as needed. Preparations to replace motor bearings should be made and replaced at next scheduled convenience (Priority 2 for motor bearings). Since excessive fan speed vibrations on the motor are likely the cause for the early deterioration of the motor bearings, the fan sheave run-out and fan/motor foundation looseness problems should be corrected before the motor bearings are replaced and as a Priority 1 repair ranking.	1 2 3
COND WTR VPMP25					●													①	3A 1A 3N	.537i .288i 2.63 g/SE	Plots 4 and 5 show harmonics of the inner race defect frequency of a SKF 7313 bearing which is used in this pump as the thrust bearing (T2). Recommend replacing both the up thrust and thrust bearings. As per Ralph Buskirk this will be done during the Christmas 1992 shutdown.	4 5
CAFETER AHU-308							●	●						●				2	1V 2BF	.786i .171i	Moderate 1X motor RPM noted at motor outboard vertical position likely due to loose/weak motor mounting system. Also, Plot 7 shows possible belt pass frequency and harmonics at Position 2 horizontal likely due to loose, worn, or mismatched belts. The sudden increase at the outboard motor position (Position 1V) may be a result of a loosened mounting bolt or broken structure. Inspect for above and make any corrections.	6 7

TAKE IMMEDIATE REPAIR ACTION DUE TO SEVERE PROBLEM(S) NOTED (SEE "RECOMMENDATIONS/REMARKS" FOR THIS MACHINE)
SCHEDULE REPAIRS TO CORRECT PROBLEMS NOTED AT CONVENIENCE. (SEE "RECOMMENDATIONS/REMARKS" FOR THIS MACHINE)
DESIGNATED PROBLEM DOES NOT WARRANT ACTION AT THIS TIME. (CONTINUE ONLY WITH RELIABILITY SURVEY TREND ANALYSIS)

©COPYRIGHT 1990

TECHNICAL ASSOCIATES OF CHARLOTTE, P.C.
(704) 333-9011
FAX (704) 333-1728

FIGURE 8

© Copyright 2001 Technical Associates Of Charlotte, P.C.

FIGURE 9 DESCRIPTION
PMP MACHINE REPAIR LOG

This Repair Log is to be completed by the mechanics who make any changes to the machinery being monitored in a PMP program. It is essential that all maintenance actions be recorded on this sheet regardless of whether the repair was ordered by the PMP department or the maintenance department performed alterations to a machine based on orders from themselves or production personnel. The reason this report must be filled out for any machine change is that the vibration signature will change and possibly require a new baselining on which to compare all future vibration measurements. A PMP Machine Repair Log is very crucial to the PMP program as it provides information used in the following manners:

1) It tells the PMP personnel that changes have taken place on the machine and a new baseline should possibly be performed.

2) The PMP personnel should take a close look at the next set of data collected to determine if other problems have occurred due to the maintenance action taken.

3) It is crucial information to monitoring the effectiveness of a PMP program as a repaired machine should show lower vibration levels as well as reduced future maintenance costs.

4) It is also a good check to see that the proper maintenance activities were performed and that there was no confusion in the process of getting a repair order from the PMP personnel to the maintenance field personnel.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

Technical Associates
of Charlotte P.C.
Phone - (704)333-9011
Fax - (704)333-1728

PMP MACHINE REPAIR LOG

Reliability Survey __of__ Predictive Maintenance Program (Page___of___)

MACHINE	DATE	MAINTENANCE ACTION

FIGURE 9

© Copyright 2001 Technical Associates Of Charlotte, P.C.

FIGURE 10 DESCRIPTION
LATEST MEASUREMENTS REPORT

This section contains a listing of **all measurement points** on the entire machine route. The printout for this section includes:

COLUMN	TEXT	DESCRIPTION
1	(Date)	Date of Vibration Survey
2	(Point ID)	Measurement Position and Direction of Transducer (horiz, vert, or axial)
3	(Curr.)	Overall Amplitude (vibration, spike energy, temperature, RPM, etc.)
4	(Prev.)	Previous Overall Amplitude (vibration, spike energy, temperature, RPM, etc.)
5	(Units)	Units of Measurement
6	(% Change)	Percent Change Since Last Survey *
7	(Warning)	Preset Overall Alarm Levels (Warning)
8	(Critical)	Preset Overall Alarm Levels (Critical)
9	(%Alm)	Percent of Alarm
10	(Ave. OA)	Average Overall Amplitude Based on the Trend of Previously Collected Overall Data (See Fig. 29)
11	(Std Dv)	Standard Deviation from Average Overage Amplitude (See Fig. 29)
12	(PK Freq.)	Frequency with the Highest Amplitude in Spectrum
13	(PK Amp.)	Amplitude of the Peak Frequency

* This column reflects the overall vibration % change since the last survey reflecting increases in machinery mechanical deterioration (+) or reductions which may be credited to machinery problem correction (-).

© Copyright 2001 Technical Associates Of Charlotte, P.C.

Page No. 1
01/13/93

LATEST OVERALL MEAS. REPORT

DATE	POINT ID	CURR.	PREV.	UNITS	%CHG	WARN.	CRIT.	%ALM	AVE.OA	STD DV	PK FREQ	PK AMPL
MACHINE	: 102B SUPY AH-44											
12/08/92	POS RPM - O	1091	1096	RPM	0	1200	2400	91	1095.7778	14.1755	No Spec	No Spec
12/08/92	POS 1 - A	1.740	1.320	IN/S	32	0.425	0.850	409	1.6076	0.5569	1080.00	1.73312
12/08/92	POS 1-H HI BPF	0.303	0.374	g/SE	-19	0.700	1.400	43	0.4638	0.1418	6300.00	0.09309
12/08/92	POS 1 - H	0.272	0.225	IN/S	21	0.425	0.850	64	0.2731	0.0662	1080.00	0.26942
12/08/92	POS 1 - V	0.950	0.582	IN/S	63	0.425	0.850	224	0.9546	0.4798	1080.00	1.30611
12/08/92	POS 2 - A	1.510	1.290	IN/S	17	0.425	0.850	355	1.2303	0.6938	1080.00	1.38459
12/08/92	POS 2-H HI BPF	0.248	0.251	g/SE	-1	0.700	1.400	35	0.3871	0.2086	6300.00	0.05600
12/08/92	POS 2 - O	0.235	0.215	IN/S	9	0.425	0.850	55	0.2993	0.0731	1170.00	0.14035
12/08/92	POS 2BF	0.264	0.262	IN/S	1	0.425	0.850	62	0.2947	0.0649	1170.00	0.18888
12/08/92	POS 2 - H	0.314	0.266	IN/S	18	0.425	0.850	74	0.3063	0.0763	1080.00	0.19722
12/08/92	POS 2 - V	0.710	0.870	IN/S	-18	0.425	0.850	167	0.6853	0.1626	1800.00	0.54504
12/08/92	POS 3-H HI BPF	0.432	0.482	g/SE	-10	0.450	0.900	96	0.4122	0.1261	150.00	0.62716
12/08/92	POS 3 - H	0.594	0.480	IN/S	24	0.425	0.850	140	0.7110	0.2261	1080.00	0.67434
12/08/92	POS 3BF	0.740	0.410	IN/S	80	0.425	0.850	174	0.6103	0.1388	1095.00	0.55536
12/08/92	POS 3 - V	0.305	0.735	IN/S	-59	0.425	0.850	72	0.8143	0.2915	1080.00	0.14932
MACHINE	: CAFETER AHU-308											
12/08/92	POS RPM - O	1255	1257	RPM	0	1400	2800	90	1266.5556	8.5602	No Spec	No Spec
12/08/92	POS 1 - A	0.052	0.039	g/SE	33	0.700	1.400	7	0.0402	0.0178	No Spec	No Spec
12/08/92	POS 1 - A	0.222	0.104	IN/S	113	0.425	0.850	52	0.1994	0.0898	1800.00	0.20278
12/08/92	POS 1 - H	0.315	0.260	IN/S	21	0.425	0.850	74	0.3155	0.0676	1800.00	0.24694
12/08/92	POS 1 - V	0.786	0.476	IN/S	65	0.425	0.850	185	0.6031	0.1792	1800.00	0.71374
12/08/92	POS 2 - A	0.035	0.017	g/SE	106	0.700	1.400	5	0.0305	0.0158	No Spec	No Spec
12/08/92	POS 2 - A	0.182	0.365	IN/S	-50	0.425	0.850	43	0.2372	0.0873	1800.00	0.15695
12/08/92	POS 2 - O	0.160	0.185	IN/S	-14	0.425	0.850	38	0.2090	0.0564	1260.00	0.09801
12/08/92	POS 2BF	0.171	0.189	IN/S	-10	0.425	0.850	40	0.1803	0.0113	315.00	0.09665
12/08/92	POS 2 - H	0.151	0.205	IN/S	-26	0.425	0.850	36	0.2109	0.0650	2520.00	0.09102
12/08/92	POS 2 - V	0.165	0.229	IN/S	-28	0.425	0.850	39	0.2134	0.0476	1800.00	0.11774
12/08/92	POS 3 - A	0.064	0.060	g/SE	7	0.500	1.000	13	0.0941	0.0775	No Spec	No Spec
12/08/92	POS 3 - A	0.247	0.198	IN/S	25	0.425	0.850	58	0.2045	0.0455	2550.00	0.16704
12/08/92	POS 3 - H	0.162	0.184	IN/S	-12	0.425	0.850	38	0.1775	0.0431	1200.00	0.09385
12/08/92	POS 3BF	0.139	0.178	IN/S	-22	0.425	0.850	33	0.1598	0.0173	1260.00	0.10790
12/08/92	POS 3 - V	0.166	0.165	IN/S	1	0.425	0.850	39	0.1647	0.0187	1200.00	0.09484
12/08/92	POS 4 - A	0.076	0.105	g/SE	-28	0.500	1.000	15	0.0976	0.0485	No Spec	No Spec
12/08/92	POS 4 - A	0.193	0.131	IN/S	47	0.425	0.850	45	0.2063	0.0543	1800.00	0.14273
12/08/92	POS 4 - H	0.185	0.131	IN/S	41	0.425	0.850	44	0.2304	0.0907	1200.00	0.11171
12/08/92	POS 4 - V	0.141	0.112	IN/S	26	0.425	0.850	33	0.1513	0.0336	1800.00	0.08060
MACHINE	: COND WTR VPMP25											
12/08/92	POS 1A - A	0.288	0.178	IN/S	62	0.200	0.400	144	0.0942	0.0559	25350.0	0.14861
12/08/92	POS 1W - O	0.163	0.098	IN/S	66	0.350	0.700	47	0.1182	0.0677	1800.00	0.03711
12/08/92	POS 1N - O	0.136	0.182	IN/S	-25	0.350	0.700	39	0.1215	0.0610	1800.00	0.10945
12/08/92	POS 1N - O	0.409	0.195	g/SE	110	0.700	1.400	58	0.1526	0.0879	No Spec	No Spec
12/08/92	POS 1EO - O	0.146	0.205	IN/S	-29	0.350	0.700	42	0.1318	0.0519	78000.0	0.06345
12/08/92	POS 2A - A	0.211	0.129	IN/S	64	0.200	0.400	106	0.0765	0.0414	12750.0	0.07728
12/08/92	POS 2W - O	0.164	0.137	IN/S	20	0.300	0.600	55	0.1009	0.0322	25350.0	0.03320
12/08/92	POS 2N - O	0.226	0.186	IN/S	22	0.300	0.600	75	0.1009	0.0476	49050.0	0.06696
12/08/92	POS 2N - O	0.327	0.539	g/SE	-39	0.700	1.400	47	0.1865	0.1284	No Spec	No Spec
12/08/92	POS 2EO - O	0.251	0.144	IN/S	74	0.300	0.600	84	0.1162	0.0527	1800.00	0.19039
12/08/92	POS 3A - A	0.537	0.421	IN/S	28	0.200	0.400	269	0.1867	0.1365	49050.0	0.34154
12/08/92	POS 3W - O	0.433	0.228	IN/S	90	0.300	0.600	144	0.1341	0.0971	49050.0	0.04575
12/08/92	POS 3N - O	0.333	0.179	IN/S	86	0.300	0.600	111	0.1139	0.0847	39900.0	0.04956
12/08/92	POS 3N - O	2.630	1.220	g/SE	116	0.700	1.400	376	0.8009	0.7211	No Spec	No Spec

FIGURE 10

© Copyright 2001 Technical Associates Of Charlotte, P.C.

FIGURE 11 DESCRIPTION
ALARM REPORTS

The **OVERALL ALARM EXCEPTION REPORT** detects all of the machines that have vibration which has exceeded the Overall Alarm for one or more points (not necessarily have exceeded the spectral alarms). These machines are often in need of corrective maintenance and are therefore of primary importance. Many of the machines may have vibration well in excess of the predetermined alarm levels on an initial survey and these need particular attention.

On the following pages are computer printouts of the **OVERALL ALARM EXCEPTION REPORT**. These columns are:

COLUMN	TEXT	DESCRIPTION
1	(Date)	Measurement Date
2	(Point ID)	Position Number Where Measurement was Taken and Direction of Transducer - Either Axial (A), Horizontal (H), or Vertical (V)
3	(Curr.)	Overall Amplitude of Current Vibration
4	(Prev.)	Overall Amplitude of Previous Vibration
5	(Units)	Units of Vibration, whether in Velocity (In/Sec), Displacement (Mils), Acceleration (g), Spike Energy (gSE), Temperature (°F),etc.
6	(Warn.)	Preset Overall Alarm 1 Levels (Warning)
7	(Crit.)	Preset Overall Alarm 2 Levels (Critical)
8	(%Alm)	Percentage of Overall Warning Alarm (100% = At Alarm Level)
9	(% Chg)	Percentage of Change since Previous Measurement
10	(Ave. OA)	Average Overall Amplitude Based on the Trend of all Previously Collected Overall Data Stored to Date (See Fig. 29)
11	(Std Dv)	Standard Deviation from Average Overall Amplitude (See Fig.29)
12	(PK Freq.)	Frequency with the Highest Amplitude in the Spectrum for this Point
13	(PK Amp.)	Amplitude of the Peak Frequency (mils, in/sec, g, gSE, etc.)

© Copyright 2001 Technical Associates Of Charlotte, P.C.

Page No. 1
01/13/93

OVERALL ALARM EXCEPTION REPORT

DATE	POINT ID	CURR.	PREV.	UNITS	WARN.	CRIT.	%ALM	%CHG	AVE.OA	STD DV	PK FREQ	PK AMPL
MACHINE : 102B SUPY AH-44												
12/08/92	POS 1 - A	1.740	1.320	IN/S	0.425	0.850	409	32	1.6076	0.5569	1080.00	1.73312
12/08/92	POS 1 - V	0.950	0.582	IN/S	0.425	0.850	224	63	0.9546	0.4798	1080.00	1.30611
12/08/92	POS 2 - A	1.510	1.290	IN/S	0.425	0.850	355	17	1.2303	0.6938	1080.00	1.38459
12/08/92	POS 2 - V	0.710	0.870	IN/S	0.425	0.850	167	-18	0.6853	0.1626	1800.00	0.54504
12/08/92	POS 3 - H	0.594	0.480	IN/S	0.425	0.850	140	24	0.7110	0.2261	1080.00	0.67434
12/08/92	POS 3BF	0.740	0.410	IN/S	0.425	0.850	174	80	0.6103	0.1388	1095.00	0.55536
MACHINE : CAFETER AHU-308												
12/08/92	POS 1 - V	0.786	0.476	IN/S	0.425	0.850	185	65	0.6031	0.1792	1800.00	0.71374
MACHINE : COND WTR VPMP25												
12/08/92	POS 1A - A	0.288	0.178	IN/S	0.200	0.400	144	62	0.0942	0.0559	25350.0	0.14861
12/08/92	POS 2A - A	0.211	0.129	IN/S	0.200	0.400	106	64	0.0765	0.0414	12750.0	0.07728
12/08/92	POS 3A - A	0.537	0.421	IN/S	0.200	0.400	269	28	0.1867	0.1365	49050.0	0.34154
12/08/92	POS 3W - O	0.433	0.228	IN/S	0.300	0.600	144	90	0.1341	0.0971	49050.0	0.04575
12/08/92	POS 3N - O	0.333	0.179	IN/S	0.300	0.600	111	86	0.1139	0.0847	39900.0	0.04956
12/08/92	POS 3N - O	2.630	1.220	g/SE	0.700	1.400	376	116	0.8009	0.7211	No Spec	No Spec

FIGURE 11

© Copyright 2001 Technical Associates Of Charlotte, P.C.

FIGURE 12 DESCRIPTION
SPECTRAL BAND ALARM REPORT

The **SPECTRAL BAND ALARM REPORT** detects all points on all PMP machines in which the vibration alarm level has been exceeded in 1 or more of the 6 spectral bands. The spectral bands have differing alarm levels in each of the 6 individual bands into which the spectrum is divided. It is important to know that these are "Power Bands" - that is, that the vibration within each band equals the RSS (Root Sum Square) level calculated for each band *. This allow higher vibration levels at 1X RPM which is normally high due to some amount of unbalance usually being present, while allowing much lower levels of vibration near bearing defect frequencies (which should not be present at all). These machines are often in need of corrective maintenance and are therefore of primary importance. Many of the machines may have vibration well in excess of the predetermined alarm levels on an initial survey and these need particular attention.

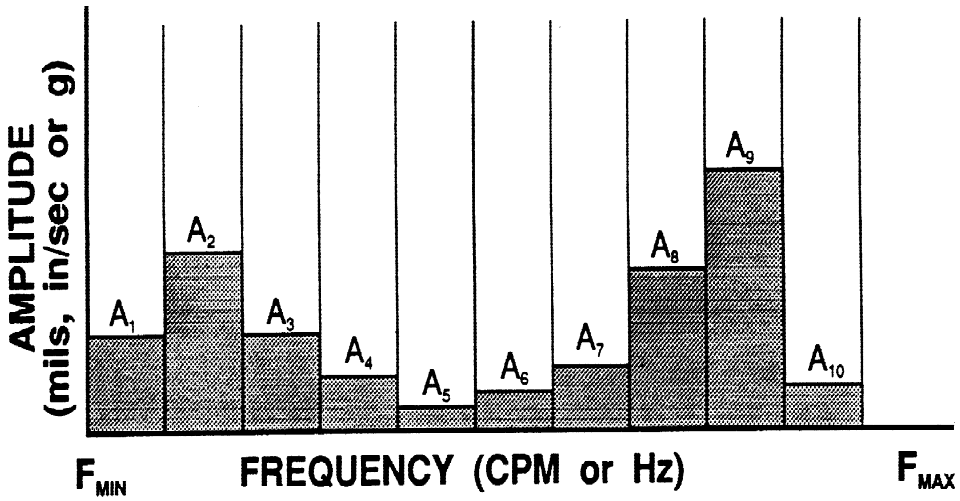
Importantly, if the RSS vibration within a band exceeds its specified alarm, an entry denoted "YES" will be printed out beside this particular band(s) in this report for each individual point listed. It is important to realize that it is not necessary for any peak within a band to exceed the band alarm - only that the total RSS value for the band equal or exceed this alarm.

Also on the following pages are computer printouts of the **SPECTRAL BAND ALARM REPORT** which has 14 columns. These 14 columns are:

COLUMN	TEXT	DESCRIPTION
1	(Date)	Measurement Date
2	(Point ID)	Position Number Where Measurement was Taken and Direction of Probe - Either Axial (A), Horizontal (H), or Vertical (V)
3	(B1)	Band 1 Amplitude
4	(ALA)	If Band Alarm is Exceeded This Column says "YES"
5	(B2)	Band 2 Amplitude
6	(ALA)	If Band Alarm is Exceeded This Column says "YES"
7	(B3)	Band 3 Amplitude
8	(ALA)	If Band Alarm is Exceeded This Column says "YES"
9	(B4)	Band 4 Amplitude
10	(ALA)	If Band Alarm is Exceeded This Column says "YES"
11	(B5)	Band 5 Amplitude
12	(ALA)	If Band Alarm is Exceeded This Column says "YES"
13	(B6)	Band 6 Amplitude
14	(ALA)	If Band Alarm is Exceeded This Column says "YES"

*NOTE: "RSS Amplitude" = Root Sum Square Amplitude (in layman's terms, it is a "summation of energy" within the band, which includes the amplitudes of each FFT line from the beginning of the band to the end of the band). This is illustrated in Illustration 1.

© Copyright 2001 Technical Associates Of Charlotte, P.C.



$$OA_{RSS} = \frac{\sqrt{\sum_{i=1}^{n} A_i^2}}{\sqrt{N_{BF}}} = \frac{\sqrt{A_1^2+A_2^2+A_3^2+....+A_n^2}}{\sqrt{1.5}}$$

OA_{RSS} = Digital Overall Level of Vibration Spectrum.

RSS = Root Sum Square Level which is the "Summation of Energy" in the spectrum from F_{MIN} to F_{MAX}.

n = Number of FFT Lines of Resolution.

A_i = Amplitude of Each of the FFT Lines.

N_{BF} = Noise Bandwidth for Window Chosen 1.5 for Hanning Window.

Which means $OA_{RSS} = .8165\sqrt{\sum_{i=1}^{n} A_i^2}$

ILLUSTRATION 1
EXACT EQUATION FOR CALCULATING DIGITAL OVERALL LEVEL OF A SPECTRUM

© Copyright 2001 Technical Associates Of Charlotte, P.C.

Page No. 1
01/05/93

BAND ALARM REPORT

DATE	POINT ID	B1	ALM	B2	ALM	B3	ALM	B4	ALM	B5	ALM	B6	ALM
MACHINE	: 102B SUPY AH-44												
12/08/92	POS 1 - A	1.8260	YES	0.0768	---	0.0123	---	0.1064	---	0.0159	---	0.0570	---
12/08/92	POS 1 - V	1.5542	YES	0.1053	---	0.0060	---	0.0152	---	0.0107	---	0.0182	---
12/08/92	POS 2 - A	1.4431	YES	0.0794	---	0.0137	---	0.0824	---	0.0118	---	0.0362	---
12/08/92	POS 2BF	0.0115	---	0.1838	---	0.1449	YES	0.0773	---	0.0376	---	0.0328	---
12/08/92	POS 2 - V	0.6363	YES	0.0778	---	0.0235	---	0.0436	---	0.0207	---	0.0345	---
12/08/92	POS 3 - H	0.7700	YES	0.1247	---	0.0619	---	0.0629	---	0.0167	---	0.0307	---
12/08/92	POS 3BF	0.1385	YES	0.6765	YES	0.3511	YES	0.0887	---	0.0530	---	0.0864	YES
MACHINE	: CAFETER AHU-308												
12/08/92	POS 1 - V	0.7380	YES	0.0922	---	0.0731	---	0.0101	---	0.0046	---	0.0029	---
12/08/92	POS 2BF	0.1035	YES	0.1066	---	0.0085	---	0.0907	---	0.0100	---	0.0268	---
MACHINE	: COND WTR VPMP25												
12/08/92	POS 1A - A	0.0776	---	0.0060	---	0.0110	---	0.1055	YES	0.1832	YES	0.0494	YES
12/08/92	POS 1EO - O	0.0458	---	0.1099	YES	0.0194	---	0.0059	---	0.0034	---	0.0014	---
12/08/92	POS 2A - A	0.0323	---	0.0106	---	0.0075	---	0.1042	YES	0.0742	YES	0.0361	---
12/08/92	POS 2N - O	0.0522	---	0.0067	---	0.0039	---	0.0422	---	0.0583	---	0.0734	YES
12/08/92	POS 3A - A	0.0128	---	0.0078	---	0.0107	---	0.0367	---	0.0902	YES	0.5254	YES

FIGURE 12

© Copyright 2001 Technical Associates Of Charlotte, P.C.

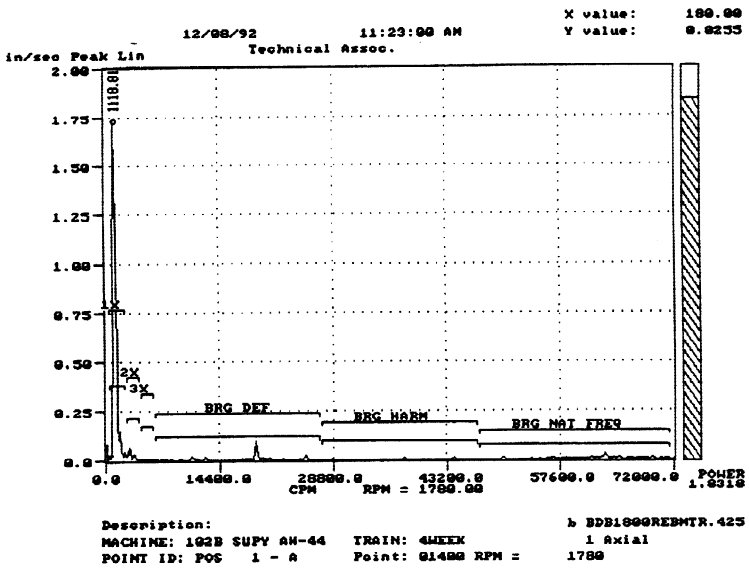
FIGURES 13 THRU 19 DESCRIPTION
GRAPHICS REPORT FOR SPECTRA HAVING SPECTRAL ALARM BANDS

A. Vibration Spectra

This section includes plots of Amplitude versus Frequency (with Spectral Alarm Bands displayed on them) which were referenced in **"RANK-ORDERED RESULTS AND RECOMMENDATIONS REPORT"** (Figure 8). Spectra are included for those machines which have either experienced significant vibration increases, are now in **ALARM**, or which have individual frequencies indicating duress of a particular machine component(s) whether or not the amplitude of these particular frequencies have increased overall levels into **ALARM** severity. Where applicable, additional notation is provided on the plot relating information on problem components and/or conditions which have been detected. Machine name, point location, vibration parameter, amplitude scale and frequency range are provided on each plot, along with spectral band alarms.

Figures 13 thru 19 (Plots 1 thru 7) are the plots of the spectra from the **RANK-ORDERED SUMMARY OF RESULTS AND RECOMMENDATIONS FOR PROBLEM MACHINES** (See Figure 8) that have been diagnosed as having problems. Note that the "Plot Nos." have been referred to in the last column of this report so that the reader can compare the diagnosed problem to the spectra from which the diagnosis was made.

© Copyright 2001 Technical Associates Of Charlotte, P.C.



12/08/92
11:23:00 AM

Threshold Report

PLANT:
Description:
MACHINE: 102B SUPY AH-44
POINT ID: POS 1 - A
Units: in/sec Peak Lin
Threshold: 0.0100

TRAIN: 4WEEK
Point: 01400
RPM: 1780.00

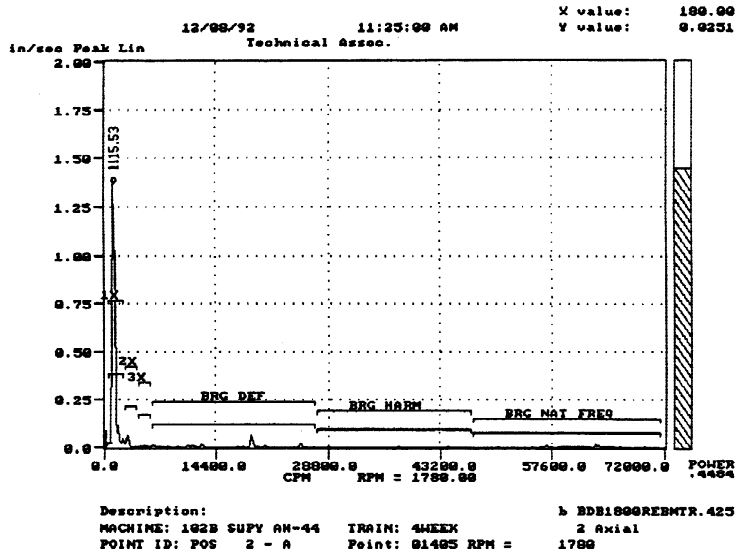
Frequency Hz	CPM	Order	Spect Amp	% Exceeded
3.00	180	0.10	0.0255	154.53
18.65	1.12E+3	0.63	1.7331	17231.20
48.97	2938	1.65	0.0629	528.84
58.62	3517	1.98	0.0279	178.62
180.98	1.09E+4	6.10	0.0200	99.98
210.00	12600	7.08	0.0157	57.12
314.91	1.89E+4	10.62	0.0959	859.07
330.57	19834	11.14	0.0103	3.02
343.96	2.06E+4	11.59	0.0103	3.02
419.91	2.52E+4	14.15	0.0247	146.97
944.56	56673	31.84	0.0103	3.02
1038.36	6.23E+4	35.00	0.0120	19.78
1056.77	6.34E+4	35.62	0.0305	2.05E+2
1074.95	64497	36.23	0.0103	3.02
1155.23	6.93E+4	38.94	0.0103	3.02

Power	Sync	Subsync	Non-Sync
1.8318	0.1592	1.8194	0.1404

PLOT 1

FIGURE 13

© Copyright 2001 Technical Associates Of Charlotte, P.C.



12/08/92
11:25:00 AM

Threshold Report

PLANT:
Description:
MACHINE: 102B SUPY AH-44
TRAIN: 4WEEK
POINT ID: POS 2 - A
Point: 01405
Units: in/sec Peak Lin
RPM: 1780.00
Threshold: 0.0100

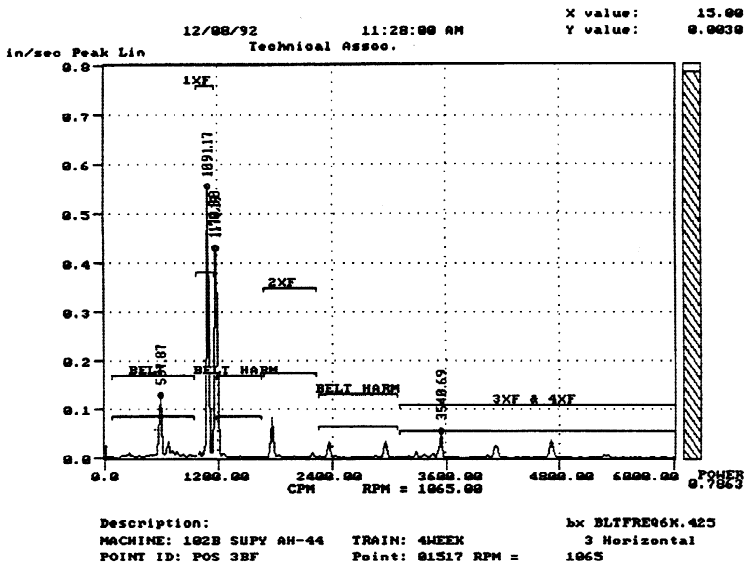
Frequency Hz	CPM	Order	Spect Amp	% Exceeded
3.00	180	0.10	0.0251	151.29
18.59	1.12E+3	0.63	1.3846	13745.90
39.32	2359	1.33	0.0485	384.85
49.24	2954	1.66	0.0673	573.48
105.91	6.35E+3	3.57	0.0130	30.24
181.16	10869	6.11	0.0122	21.95
190.60	11436	6.42	0.0107	6.93
210.70	1.26E+4	7.10	0.0206	106.32
315.94	18956	10.65	0.0652	551.71
421.25	25275	14.20	0.0228	127.70
947.42	56845	31.94	0.0122	21.95
1052.74	6.32E+4	35.49	0.0153	53.50

Power	Sync	Subsync	Non-Sync
1.4484	0.1202	1.4380	0.1241

PLOT 2

FIGURE 14

© Copyright 2001 Technical Associates Of Charlotte, P.C.



12/08/92
11:28:00 AM

Threshold Report

PLANT:
Description:
MACHINE: 102B SUPY AH-44 TRAIN: 4WEEK
POINT ID: POS 3BF Point: 01517
Units: in/sec Peak Lin RPM: 1065.00
Threshold: 0.0100

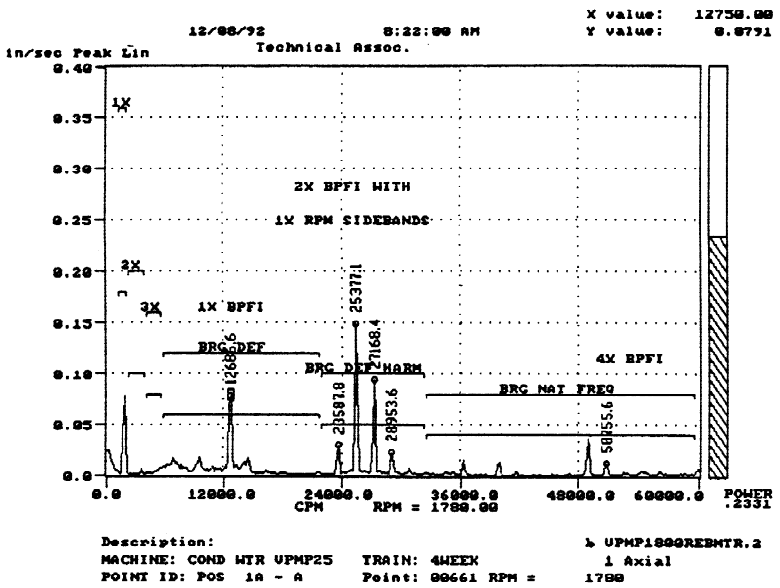
Frequency Hz	CPM	Order	Spect Amp	% Exceeded
4.20	2.52E+2	0.24	0.0112	12.00
9.80	5.88E+2	0.55	0.1275	1174.51
11.17	670	0.63	0.0322	221.96
11.92	7.15E+2	0.67	0.0149	49.38
12.77	766	0.72	0.0136	35.71
16.72	1.00E+3	0.94	0.0149	49.38
18.19	1091	1.02	0.5554	5453.61
19.62	1177	1.11	0.4299	4199.37
20.85	1251	1.17	0.0102	1.75
29.27	1756	1.65	0.0841	740.80
36.39	2183	2.05	0.0108	8.47
39.31	2.36E+3	2.21	0.0332	232.43
49.15	2949	2.77	0.0332	232.43
54.59	3.28E+3	3.08	0.0140	40.12
59.01	3.54E+3	3.32	0.0555	454.67
68.65	4.12E+3	3.87	0.0266	165.72
78.47	4708	4.42	0.0366	265.92

Power	Sync	Subsync	Non-Sync
0.7863	0.3101	0.1394	0.7090

PLOT 3

FIGURE 15

© Copyright 2001 Technical Associates Of Charlotte, P.C.



12/08/92
8:22:00 AM

Threshold Report

PLANT:
Description:
MACHINE: COND WTR VPMP25 TRAIN: 4WEEK
POINT ID: POS 1A - A Point: 00661
Units: in/sec Peak Lin RPM: 1780.00
Threshold: 0.0100

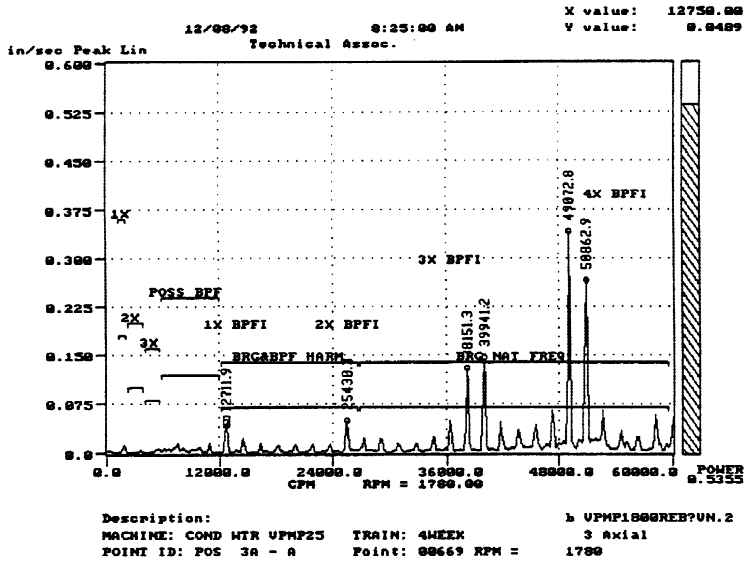
Frequency Hz	CPM	Order	Spect Amp	% Exceeded
2.50	150	0.08	0.0241	141.48
29.88	1.79E+3	1.01	0.0771	671.03
103.65	6219	3.49	0.0105	5.00E+0
113.56	6.81E+3	3.83	0.0165	65.38
127.50	7650	4.30	0.0102	2.38
158.53	9.51E+3	5.34	0.0170	69.60
211.44	1.27E+4	7.13	0.0791	690.74
241.48	1.45E+4	8.14	0.0170	69.60
393.13	2.36E+4	13.25	0.0296	195.51
422.95	25377	14.26	0.1486	1386.10
452.81	27168	15.26	0.0944	843.53
482.56	2.90E+4	16.27	0.0224	123.87
604.34	36260	20.37	0.0153	53.32
664.11	3.98E+4	22.39	0.0128	28.49
815.98	4.90E+4	27.50	0.0362	261.62
845.93	5.08E+4	28.51	0.0119	19.12

Power	Sync	Subsync	Non-Sync
0.2332	0.1025	0.0321	0.2069

PLOT 4

FIGURE 16

© Copyright 2001 Technical Associates Of Charlotte, P.C.



12/08/92
8:25:00 AM

Threshold Report

PLANT:
Description:
MACHINE: COND WTR VPMP25
POINT ID: POS 3A - A
Units: in/sec Peak Lin
Threshold: 0.0200
TRAIN: 4WEEK
Point: 00669
RPM: 1780.00

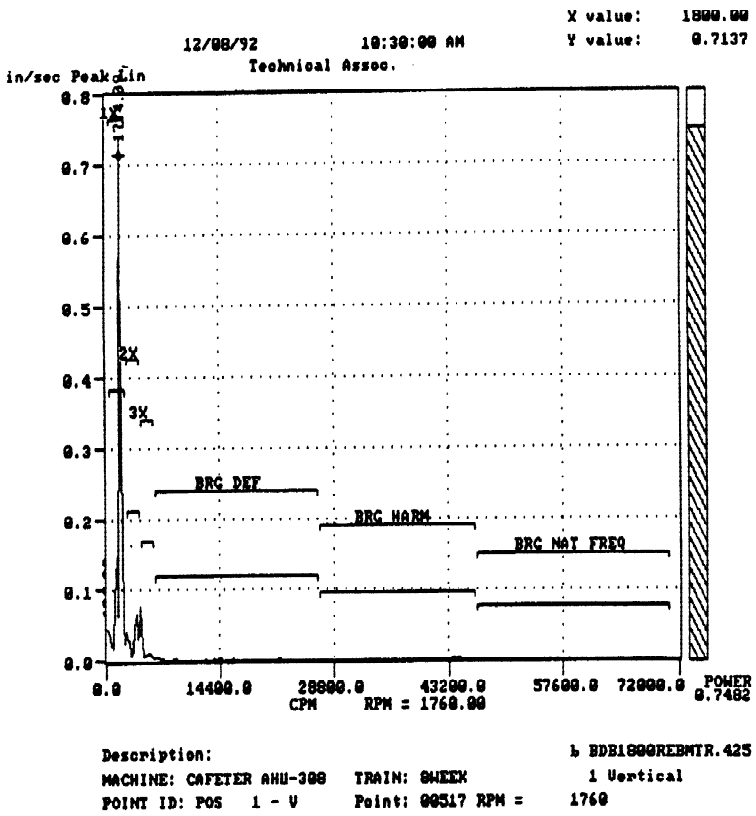
Frequency Hz	CPM	Order	Spect Amp	% Exceeded
211.87	1.27E+4	7.14	0.0489	144.55
241.73	1.45E+4	8.15	0.0234	16.79
423.84	25430	14.29	0.0503	151.34
453.65	2.72E+4	15.29	0.0240	20.03
483.66	29019	16.30	0.0234	16.79
575.98	3.46E+4	19.42	0.0254	26.78
605.85	36351	20.42	0.0503	151.34
635.86	38151	21.43	0.1310	555.14
665.69	39941	22.44	0.1462	630.95
695.64	4.17E+4	23.45	0.0476	137.95
726.37	43582	24.48	0.0372	85.99
757.37	45442	25.53	0.0438	119.19
787.90	4.73E+4	26.56	0.0661	230.48
807.36	48441	27.21	0.0204	1.85
817.88	4.91E+4	27.57	0.3415	1607.70
835.82	50149	28.17	0.0209	4.68
847.72	5.09E+4	28.57	0.2670	1234.82
864.32	5.19E+4	29.13	0.0240	20.03
877.88	5.27E+4	29.59	0.0661	230.48
909.29	5.46E+4	30.65	0.0362	80.97
938.67	56320	31.64	0.0254	26.78
969.96	5.82E+4	32.70	0.0592	196.20

Power	Sync	Subsync	Non-Sync
0.5373	0.0711	0.0056	0.5326

PLOT 5

FIGURE 17

© Copyright 2001 Technical Associates Of Charlotte, P.C.



12/08/92
10:30:00 AM

Threshold Report

PLANT:
Description:
MACHINE: CAFETER AHU-308
POINT ID: POS 1 - V
Units: in/sec Peak Lin
Threshold: 0.0100

TRAIN: 8WEEK
Point: 00517
RPM: 1760.00

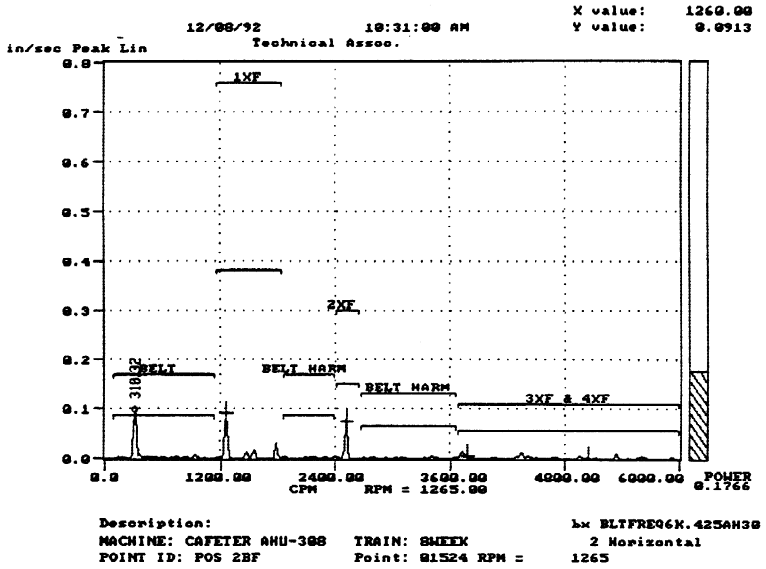
Frequency Hz	CPM	Order	Spect Amp	% Exceeded
3.00	180	0.10	0.0419	318.51
29.57	1774	1.01	0.7137	7037.40
62.27	3736	2.12	0.0664	564.10
72.21	4.33E+3	2.46	0.0758	657.76

Power Sync Subsync Non-Sync

PLOT 6

FIGURE 18

© Copyright 2001 Technical Associates Of Charlotte, P.C.



12/08/92
10:31:00 AM

Threshold Report

PLANT:
Description:
MACHINE: CAFETER AHU-308 TRAIN: 8WEEK
POINT ID: POS 2BF Point: 01524
Units: in/sec Peak Lin RPM: 1265.00
Threshold: 0.0100

Frequency Hz	CPM	Order	Spect Amp	% Exceeded
5.17	310	0.25	0.0966	866.48
20.91	1.25E+3	0.99	0.0913	812.51
24.43	1.47E+3	1.16	0.0112	12.01
25.81	1.55E+3	1.22	0.0167	67.49
29.64	1.78E+3	1.41	0.0315	215.15
41.87	2.51E+3	1.99	0.0790	690.39
61.94	3716	2.94	0.0137	36.97
72.28	4337	3.43	0.0109	8.84

Power	Sync	Subsync	Non-Sync
0.1766	0.1159	0.1039	0.0834

PLOT 7

FIGURE 19

© Copyright 2001 Technical Associates Of Charlotte, P.C.

FIGURE 20 DESCRIPTION
SPECTRAL NARROWBAND ENVELOPE ALARM REPORT

The **SPECTRAL NARROWBAND ENVELOPE ALARM REPORT** only prints out spectra for those points in which at least one peak has equaled or exceeded a narrowband envelope alarm. Narrowband envelope alarms are established by the PMP software based on input of the analyst after at least one set of spectra have been captured for a machine. Currently, some PMP software requires a machine to be in good operating condition when these are set. In these types of software systems, the narrowband alarms are then calculated by setting the envelope a certain percentage above each frequency (for example, setting the envelope 50% to 100% above the amplitude of each peak in the baseline spectrum when the machine is in good operating condition).

Other software offerings allow the user to take in large numbers of similar machines into a "family" and simultaneously create narrowband alarms for the whole machine "family". Of course, some machines will likely be in very good condition, some in very bad shape and the remaining in between these extremes. Therefore, these types of software systems allow the user to go ahead and group large numbers of machines into such a "family" and then employ statistical analysis to have the computer automatically create the envelopes. In these cases this is done by having the software compute the average level of each peak on each spectra for every survey preformed, and for every point on each PMP machine. Of course, if the user desires, he can have the software create a different envelope alarm for each position and direction. That is, one envelope setup could be created for position 1A (axial) on each outboard motor bearing in the "family"; then another envelope alarm for position 1H (horizontal) on each machine, etc. Figure 21 through 27 following the **SPECTRAL NARROWBAND ALARM REPORT** are example spectra having at least one envelope in alarm (these spectra are the same Plots 1 thru 7 to which the **RANK-ORDERED RESULTS AND RECOMMENDATIONS REPORT** of Figure 8 referred).

The following columns appear on the report with an explanation of their contents:

COLUMN	TEXT	DESCRIPTION
1	(Date)	Measurement Date
2	(Point ID)	Position Number Where Measurement was Taken and Direction of Probe - Either Axial (A), Horizontal (H), or Vertical (V).
3	(N'Band ALM-1)	Narrowband Alarm Name that was used to determine if an Alarm has occurred.
4	(Over)	"YES" in this column signifies that at least one Narrowband Envelope Alarm was exceeded.
5	(PK Freq.)	Frequency with the Highest Amplitude in Spectrum for this Point
6	(PK Amp.)	Amplitude of the Peak Frequency (mils, in/sec, g, gSE, etc.)
7	(Units)	Units of Vibration, whether in Velocity (In/Sec), Displacement (Mils), Acceleration (g), Spike Energy (gSE).
8	(OA)	Amplitude of Current Overall Vibration Measurement *.
9	(OA/ALM)	Predetermined Overall Alarm Level
10	(% ALM	Percentage of Overall Warning Alarm (100% = At Alarm Level)
11	(% CHG)	Percentage Change of Overall since Previous Survey.
12	(F_{MAX})	Maximum Frequency Specified when the Data was Collected
13	(Point)	Point Number assigned by the PMP Software to that Particular Measurement Point

* - Overall vibration level here is calculated from the time waveform collected at the time of measurement. In this data collection used, it includes the frequency range of 300 to 3,900,000 CPM (5 to 65,000 Hz) independent of the frequency span specified. It is important to note that this is not the value which has been calculated from the spectral data collected which would only include vibration data from 0 CPM to F_{MAX} at the time of date collection.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

Page No. 1
01/15/93

SPECTRAL NARROWBAND ALARM REPORT

DATE	POINT ID	N'BAND ALM-1	OVER	PK FREQ	PK AMPL	UNITS	OA	OA/ALM	%ALM	%CHG	FMAX	POINT
MACHINE	: 102B SUPY AH-44											
12/08/92	POS 1 - A	WARNINGLIST	YES	1080.00	1.73312	IN/S	1.740	0.4250	409	32	72000.0	01400
12/08/92	POS 1 - V	WARNINGLIST	YES	1080.00	1.30611	IN/S	0.950	0.4250	224	63	72000.0	01404
12/08/92	POS 2 - A	WARNINGLIST	YES	1080.00	1.38459	IN/S	1.510	0.4250	355	17	72000.0	01405
12/08/92	POS 2 - V	WARNINGLIST	YES	1800.00	0.54504	IN/S	0.710	0.4250	167	-18	72000.0	01409
12/08/92	POS 3 - H	WARNINGLIST	YES	1080.00	0.67434	IN/S	0.594	0.4250	140	24	54000.0	01411
12/08/92	POS 3BF	WARNINGLIST	YES	1095.00	0.55536	IN/S	0.740	0.4250	174	80	6000.00	01517
MACHINE	: CAFETER AHU-308											
12/08/92	POS 1 - H	WARNINGLIST	YES	1800.00	0.24694	IN/S	0.315	0.4250	74	21	72000.0	00515
12/08/92	POS 1 - V	WARNINGLIST	YES	1800.00	0.71374	IN/S	0.786	0.4250	185	65	72000.0	00517
12/08/92	POS 2BF	WARNINGLIST	YES	315.00	0.09665	IN/S	0.171	0.4250	40	-10	6000.00	01524
MACHINE	: COND WTR VPMP25											
12/08/92	POS 1A - A	WARNINGLIST	YES	25350.0	0.14861	IN/S	0.288	0.2000	144	62	60000.0	00661
12/08/92	POS 3A - A	WARNINGLIST	YES	49050.0	0.34154	IN/S	0.537	0.2000	269	28	60000.0	00669

FIGURE 20

© Copyright 2001 Technical Associates Of Charlotte, P.C.

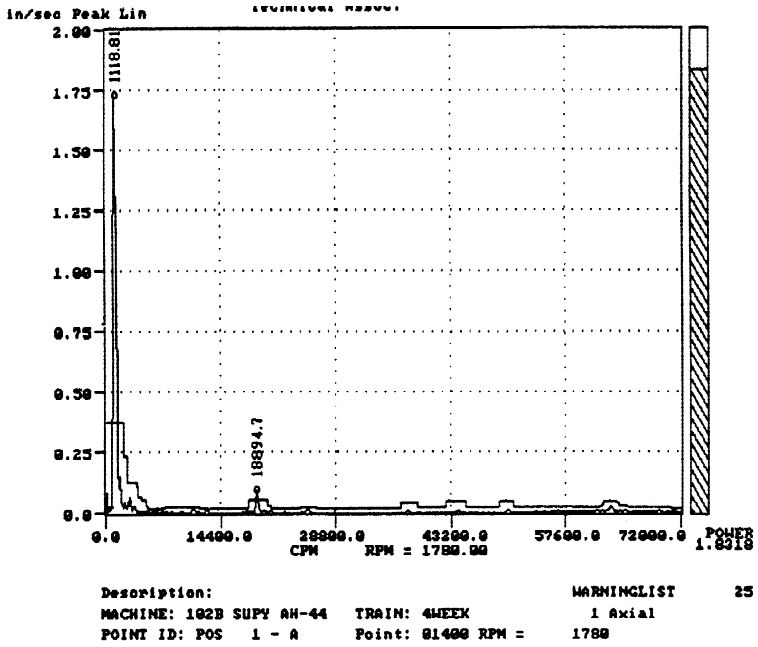
FIGURES 21 THRU 27 DESCRIPTION
GRAPHICS REPORT FOR SPECTRA HAVING NARROWBAND ENVELOPE ALARMS

A. Vibration Spectra

This section includes plots of Vibration Amplitude versus Frequency (with Narrowband Spectral Envelope Bands displayed on them). Spectra are included for those machines which have either experienced significant vibration increases, are now in **ALARM** as determined by the **NARROWBAND ALARM REPORT**, or which have individual frequencies indicating duress of a particular machine component(s) whether or not the amplitude of these particular frequencies have increased overall levels into **ALARM** severity. Where applicable, additional notation is provided on the plot relating information on problem components and/or conditions which have been detected. Machine name, point location, vibration parameter, amplitude scale and frequency range are provided on each plot, along with the narrowband envelope alarms.

Figures 21 thru 27 (Plots 1 thru 7) are the plots of the spectra from the **RANK-ORDERED SUMMARY OF RESULTS AND RECOMMENDATIONS FOR PROBLEM MACHINES** (See Figure 8) which were machines having been diagnosed as having problems. Note that the "Plot Nos." were referred to in the last column of this report (See Figure 8) so that the reader can compare the diagnosed problem to the spectra from which the diagnosis was made.

© Copyright 2001 Technical Associates Of Charlotte, P.C.



12/08/92
11:23:00 AM

Threshold Report

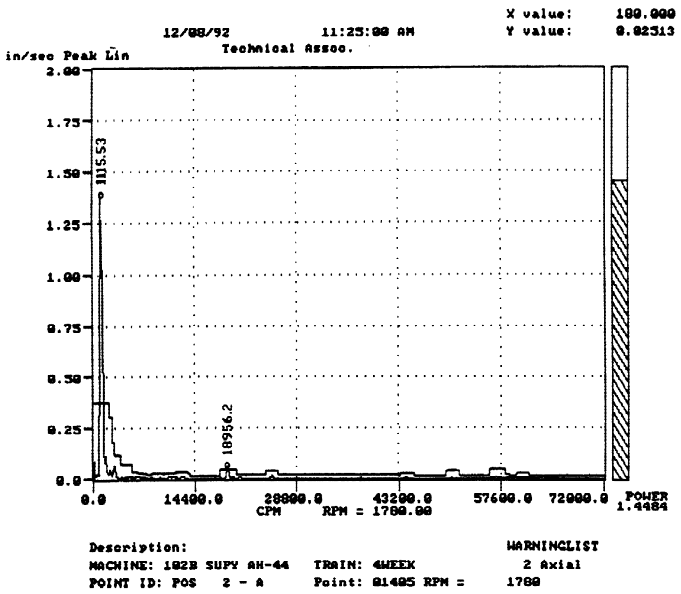
PLANT:
Description:
MACHINE: 102B SUPY AH-44 TRAIN: 4WEEK
POINT ID: POS 1 - A Point: 01400
Units: in/sec Peak Lin RPM: 1780.00
Threshold: 0.0100

Frequency Hz	CPM	Order	Spect Amp	% Exceeded
3.00	180	0.10	0.0255	154.53
18.65	1.12E+3	0.63	1.7331	17231.20
48.97	2938	1.65	0.0629	528.84
58.62	3517	1.98	0.0279	178.62
180.98	1.09E+4	6.10	0.0200	99.98
210.00	12600	7.08	0.0157	57.12
314.91	1.89E+4	10.62	0.0959	859.07
330.57	19834	11.14	0.0103	3.02
343.96	2.06E+4	11.59	0.0103	3.02
419.91	2.52E+4	14.15	0.0247	146.97
944.56	56673	31.84	0.0103	3.02
1038.36	6.23E+4	35.00	0.0120	19.78
1056.77	6.34E+4	35.62	0.0305	2.05E+2
1074.95	64497	36.23	0.0103	3.02

PLOT 1

FIGURE 21

© Copyright 2001 Technical Associates Of Charlotte, P.C.



12/08/92
11:25:00 AM

Threshold Report

PLANT:
Description:
MACHINE: 102B SUPY AH-44 TRAIN: 4WEEK
POINT ID: POS 2 - A Point: 01405
Units: in/sec Peak Lin RPM: 1780.00
Threshold: 0.0100

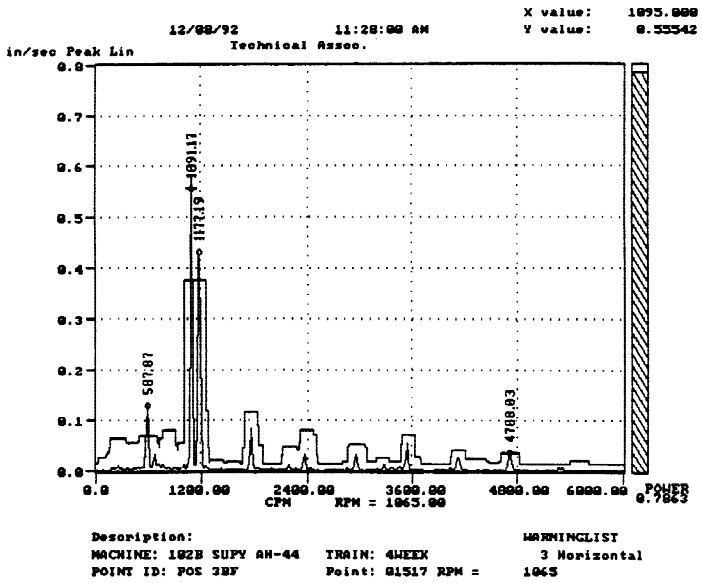
Frequency Hz	CPM	Order	Spect Amp	% Exceeded
3.00	180	0.10	0.0251	151.29
18.59	1.12E+3	0.63	1.3846	13745.90
39.32	2359	1.33	0.0485	384.85
49.24	2954	1.66	0.0673	573.48
105.91	6.35E+3	3.57	0.0130	30.24
181.16	10869	6.11	0.0122	21.95
190.60	11436	6.42	0.0107	6.93
210.70	1.26E+4	7.10	0.0206	106.32
315.94	18956	10.65	0.0652	551.71
421.25	25275	14.20	0.0228	127.70
947.42	56845	31.94	0.0122	21.95
1052.74	6.32E+4	35.49	0.0153	53.50

Power	Sync	Subsync	Non-Sync
1.4484	0.1202	1.4380	0.1241

PLOT 2

FIGURE 22

© Copyright 2001 Technical Associates Of Charlotte, P.C.



12/08/92
11:28:00 AM

Threshold Report

PLANT:
Description:
MACHINE: 102B SUPY AH-44 TRAIN: 4WEEK
POINT ID: POS 3BF Point: 01517
Units: in/sec Peak Lin RPM: 1065.00
Threshold: 0.0100

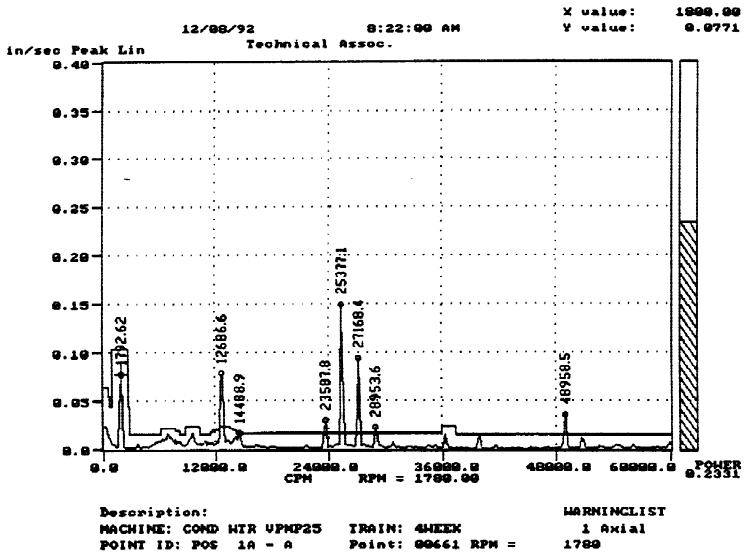
Frequency Hz	CPM	Order	Spect Amp	% Exceeded
4.20	2.52E+2	0.24	0.0112	12.00
9.80	5.88E+2	0.55	0.1275	1174.51
11.17	670	0.63	0.0322	221.96
11.92	7.15E+2	0.67	0.0149	49.38
12.77	766	0.72	0.0136	35.71
16.72	1.00E+3	0.94	0.0149	49.38
18.19	1091	1.02	0.5554	5453.61
19.62	1177	1.11	0.4299	4199.37
20.85	1251	1.17	0.0102	1.75
29.27	1756	1.65	0.0841	740.80
36.39	2183	2.05	0.0108	8.47
39.31	2.36E+3	2.21	0.0332	232.43
49.15	2949	2.77	0.0332	232.43
54.59	3.28E+3	3.08	0.0140	40.12
59.01	3.54E+3	3.32	0.0555	454.67
68.65	4.12E+3	3.87	0.0266	165.72
78.47	4708	4.42	0.0366	265.92

Power	Sync	Subsync	Non-Sync
0.7863	0.3101	0.1394	0.7090

PLOT 3

FIGURE 23

© Copyright 2001 Technical Associates Of Charlotte, P.C.



12/08/92
8:22:00 AM

Threshold Report

PLANT:
Description:
MACHINE: COND WTR VPMP25 TRAIN: 4WEEK
POINT ID: POS 1A - A Point: 00661
Units: in/sec Peak Lin RPM: 1780.00
Threshold: 0.0100

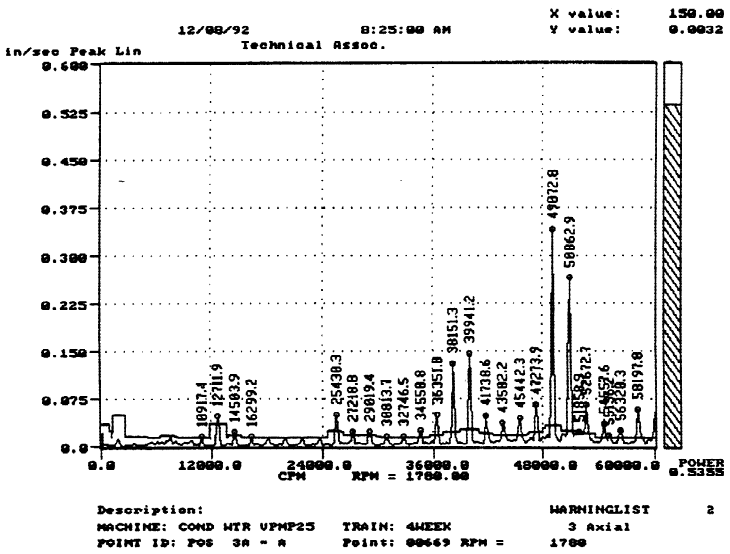
Frequency Hz	CPM	Order	Spect Amp	% Exceeded
2.50	150	0.08	0.0241	141.48
29.88	1.79E+3	1.01	0.0771	671.03
103.65	6219	3.49	0.0105	5.00E+0
113.56	6.81E+3	3.83	0.0165	65.38
127.50	7650	4.30	0.0102	2.38
158.53	9.51E+3	5.34	0.0170	69.60
211.44	1.27E+4	7.13	0.0791	690.74
241.48	1.45E+4	8.14	0.0170	69.60
393.13	2.36E+4	13.25	0.0296	195.51
422.95	25377	14.26	0.1486	1386.10
452.81	27168	15.26	0.0944	843.53
482.56	2.90E+4	16.27	0.0224	123.87
604.34	36260	20.37	0.0153	53.32
664.11	3.98E+4	22.39	0.0128	28.49
815.98	4.90E+4	27.50	0.0362	261.62
845.93	5.08E+4	28.51	0.0119	19.12

Power	Sync	Subsync	Non-Sync
0.2332	0.1025	0.0321	0.2069

PLOT 4

FIGURE 24

© Copyright 2001 Technical Associates Of Charlotte, P.C.



12/08/92
8:25:00 AM

Threshold Report

PLANT:
Description:
MACHINE: COND WTR VPMP25 TRAIN: 4WEEK
POINT ID: POS 3A - A Point: 00669
Units: in/sec Peak Lin RPM: 1780.00
Threshold: 0.0200

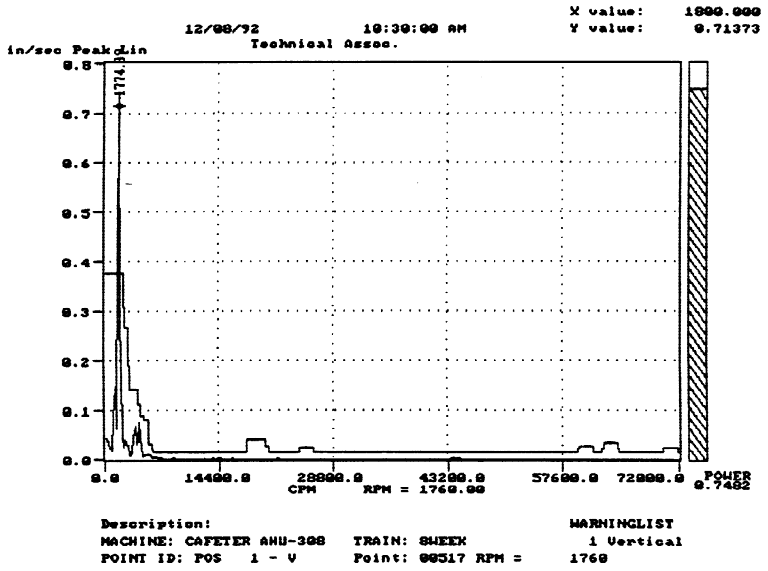
Frequency Hz	CPM	Order	Spect Amp	% Exceeded
211.87	1.27E+4	7.14	0.0489	144.55
241.73	1.45E+4	8.15	0.0234	16.79
423.84	25430	14.29	0.0503	151.34
453.65	2.72E+4	15.29	0.0240	20.03
483.66	29019	16.30	0.0234	16.79
575.98	3.46E+4	19.42	0.0254	26.78
605.85	36351	20.42	0.0503	151.34
635.86	38151	21.43	0.1310	555.14
665.69	39941	22.44	0.1462	630.95
695.64	4.17E+4	23.45	0.0476	137.95
726.37	43582	24.48	0.0372	85.99
757.37	45442	25.53	0.0438	119.19
787.90	4.73E+4	26.56	0.0661	230.48
807.36	48441	27.21	0.0204	1.85
817.88	4.91E+4	27.57	0.3415	1607.70
835.82	50149	28.17	0.0209	4.68
847.72	5.09E+4	28.57	0.2670	1234.82
864.32	5.19E+4	29.13	0.0240	20.03
877.88	5.27E+4	29.59	0.0661	230.48
909.29	5.46E+4	30.65	0.0362	80.97
938.67	56320	31.64	0.0254	26.78
969.96	5.82E+4	32.70	0.0592	196.20

Power	Sync	Subsync	Non-Sync
0.5373	0.0711	0.0056	0.5326

PLOT 5

FIGURE 25

© Copyright 2001 Technical Associates Of Charlotte, P.C.



12/08/92
10:30:00 AM

Threshold Report

PLANT:
Description:
MACHINE: CAFETER AHU-308 TRAIN: 8WEEK
POINT ID: POS 1 - V Point: 00517
Units: in/sec Peak Lin RPM: 1760.00
Threshold: 0.0100

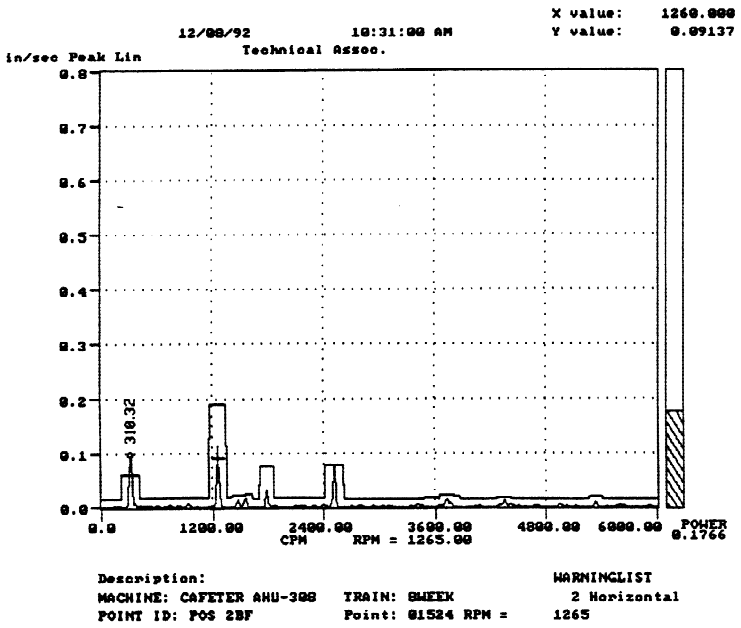
Frequency Hz	CPM	Order	Spect Amp	% Exceeded
3.00	180	0.10	0.0419	318.51
29.57	1774	1.01	0.7137	7037.40
62.27	3736	2.12	0.0664	564.10
72.21	4.33E+3	2.46	0.0758	657.76

Power	Sync	Subsync	Non-Sync
0.7482	0.7237	0.1579	0.1058

PLOT 6

FIGURE 26

© Copyright 2001 Technical Associates Of Charlotte, P.C.



12/08/92
10:31:00 AM

Threshold Report

PLANT:
Description:
MACHINE: CAFETER AHU-308 TRAIN: 8WEEK
POINT ID: POS 2BF Point: 01524
Units: in/sec Peak Lin RPM: 1265.00
Threshold: 0.0100

Frequency Hz	CPM	Order	Spect Amp	% Exceeded
5.17	310	0.25	0.0966	866.48
20.91	1.25E+3	0.99	0.0913	812.51
24.43	1.47E+3	1.16	0.0112	12.01
25.81	1.55E+3	1.22	0.0167	67.49
29.64	1.78E+3	1.41	0.0315	215.15
41.87	2.51E+3	1.99	0.0790	690.39
61.94	3716	2.94	0.0137	36.97
72.28	4337	3.43	0.0109	8.84

Power	Sync	Subsync	Non-Sync

PLOT 7

FIGURE 27

© Copyright 2001 Technical Associates Of Charlotte, P.C.

FIGURE 28 DESCRIPTION
CURRENT INSPECTION CODE REPORT

The **INSPECTION CODE REPORT** is a listing of inspection codes stored during the survey with the corresponding machine and other applicable information. These inspection codes are entered and stored in the field when abnormalities in spectra or operating conditions are noted. Although these codes will appear in both the **ACTION** and the **MACHINERY MEASUREMENT** reports, the **CURRENT INSPECTION CODE REPORT** is included for a "quick reference" to alert maintenance personnel of conditions considered abnormal as observed during the survey.

Page No. 1
01/05/93

I N S P E C T I O N C O D E R E P O R T

TE	POINT ID	INSPECTION REMARKS
MACHINE	: 102B SUPY AH-44	
01/05/93	POS 1 - H	Fluctuating Vibration
01/05/93	POS 2-H HI BPF	Belts Loose or Missing
MACHINE	: CAFETER AHU-308	
01/05/93	POS 4 - H	Safety Condition Indicated
01/05/93	POS 1 - H	Loose Parts
01/05/93	POS 3 - H	Abnormal Noise
MACHINE	: COND WTR VPMP25	
01/05/93	POS 1A - A	Leaks
01/05/93	POS 3W - O	Fluctuating Vibration
01/05/93	POS 1N - O	Hot Bearing

FIGURE 28

© Copyright 2001 Technical Associates Of Charlotte, P.C.

FIGURES 29 THRU 31 DESCRIPTION
TRENDS AND WATERFALL PLOTS

Figure 29 shows 3 different plots. The top plot is a trend of Overall Velocity starting on 4/18/91 and progressing to the current measurement at 12/8/92 with data taken on the inboard pump bearing (vertical pump). Note that it shows a marked increase in overall velocity amplitude, but that no diagnostics can be performed from this plot alone. It merely tells the reader that something is wrong since the alarm levels (shown on the plot) have been exceeded (both ALARM 1 and ALARM 2)

The middle plot of Figure 29 show a Velocity Spectrum with narrowband envelope alarms for the latest (12/8/92) measurement in orders (orders are multiples of running speed; i.e., if a machine is running at 3580 RPM, the 1st order would be at 3580 CPM, the 2nd order would be at 7160 CPM, and so on . . .). Note that the 7.14 order of running speed peak has exceeded the narrowband alarm. This 7.14X RPM frequency correlates with the Bearing Inner Race Defect Frequency Multiplier (BPFI). Also, note that many harmonics of the BPFI order have appeared (along with many IX RPM Sideband Frequencies), and are labeled in this plot. This is a spectrum indicating significant bearing wear which must be replaced immediately.

The bottom plot of Figure 29 is a trend of Overall Spike Energy (gSE) from 4/18/91 to 12/8/92 from the same pump inboard bearing. Notice how the spike energy level has been increasing over the past several measurements. Also, note that the pattern was somewhat erratic which is common with spike energy measurements. This increase in spike energy levels can be attributed to many factors which include bearing wear, lubrication problems, cavitation, rubs, etc. However, it is necessary to view the spectrum (middle of Figure 29) to determine exactly what the problem is.

The top of Figure 30 is a waterfall plot (also known as a “Cascade Plot”) on the same inboard pump bearing (again in the axial direction) that was found to have a problem in Figure 29. A waterfall plot is a date-ordered representation of the frequency spectral history for this point. Note the dates on the right-hand side of the figure. Also, note the appearance, and then the growth of inner race frequencies for this inboard pump bearing. Analysis of the bottom spectrum in Figure 30 (from the 12/8/92 data) will clearly show these bearing defect frequencies.

Figure 31 is another waterfall plot, but taken in the axial direction on the top (outboard) motor bearing housing of this vertical pump installation (a distance of approximately 6 to 7 feet from the inboard pump bearing). Note even here on the other side of the coupling on the upper motor bearing that the pump bearing problem was detected during the same 6/9/92 survey. Here again, more pump bearing inner race frequency harmonics appeared and grew until the latest survey of 12/8/92. A strong recommendation was made to replace this inboard pump bearing ASAP.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

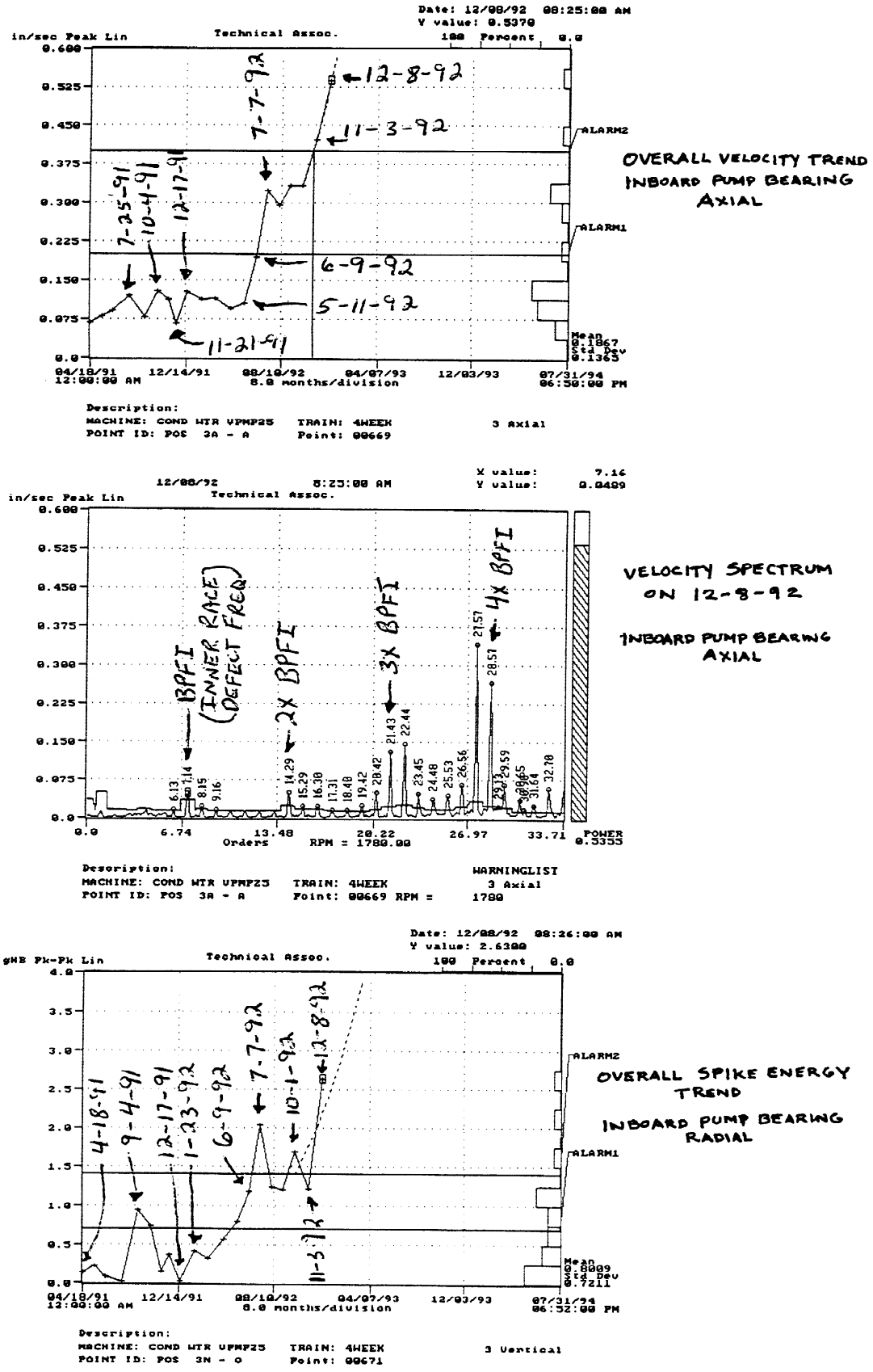


FIGURE 29

© Copyright 2001 Technical Associates Of Charlotte, P.C.

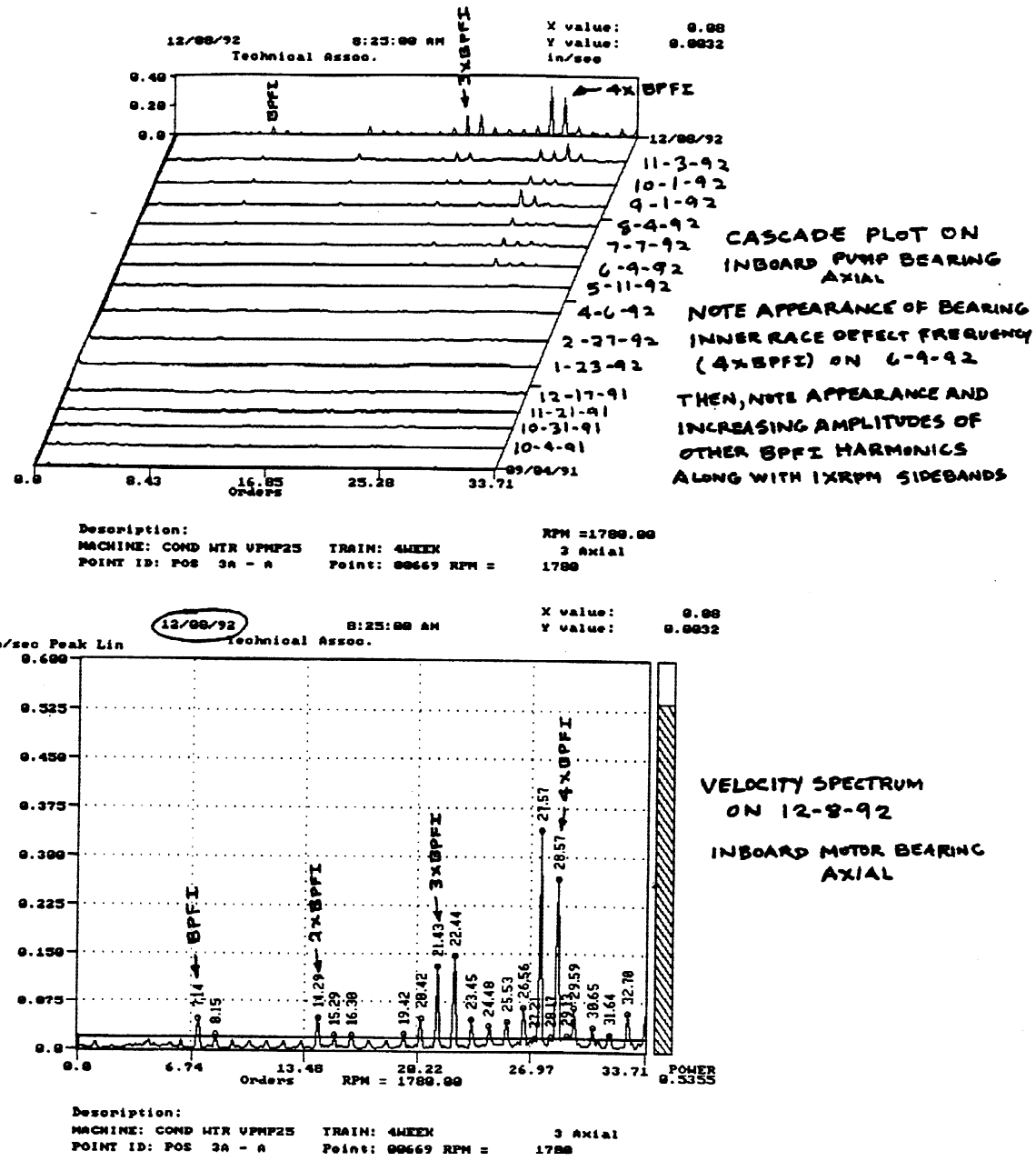
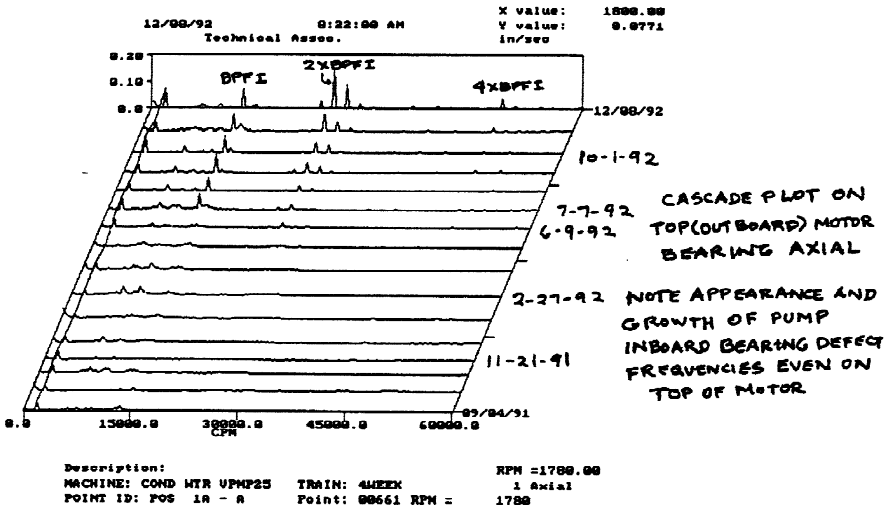


FIGURE 30

© Copyright 2001 Technical Associates Of Charlotte, P.C.



12/08/92
8:22:00 AM

Threshold Report

PLANT:
Description:
MACHINE: COND WTR VPMP25 — 12-8-92 DATA — TRAIN: 4WEEK
POINT ID: POS 1A - A — TOP MOTOR BEARING — Point: 00661
Units: in/sec Peak Lin — RPM: 1780.00
Threshold: 0.0100

Frequency Hz	CPM	Order	Spect Amp	% Exceeded
2.50	150	0.08	0.0241	141.48
29.88	1.79E+3	1.01	0.0771	671.03
103.65	6219	3.49	0.0105	5.00E+0
113.56	6.81E+3	3.83	0.0165	65.38
127.50	7650	4.30	0.0102	2.38
158.53	9.51E+3	5.34	0.0170	69.60
211.44	1.27E+4	7.13←BPFI	0.0791	690.74
241.48	1.45E+4	8.14	0.0170	69.60
393.13	2.36E+4	13.25	0.0296	195.51
422.95	25377	14.26←2BPFI	0.1486	1386.10
452.81	27168	15.26	0.0944	843.53
482.56	2.90E+4	16.27	0.0224	123.87
604.34	36260	20.37	0.0153	53.32
664.11	3.98E+4	22.39	0.0128	28.49
815.98	4.90E+4	27.50	0.0362	261.62
845.93	5.08E+4	28.51←4BPFI	0.0119	19.12

Power	Sync	Subsync	Non-Sync
0.2332	0.1025	0.0321	0.2069

FIGURE 31

© Copyright 2001 Technical Associates Of Charlotte, P.C.

EFFECT OF VIBRATION ACCEPTANCE TESTING

EFFECTS ON NEW AND REBUILT MOTOR COSTS AND MAINTENANCE CREW SIZE

(Excerpt from Practical Solution to Machinery and Maintenance Vibration Problems: Ralph T. Buscarello; Update International, Inc., 6320 West Lakeridge Road, Denver, CO 80227)

During harsh economic times, companies are often forced to maintain the same number of machines with less people. During times of expansion, they often have to add more people. But look at how vibration control can affect maintenance crew size, to save in both harsh as well as good times. A chemical plant with 1700 rotating machine assemblies driven by 1200 electric motors reported the following results:

From mid 1977 through mid 1980, they reduced their motor rewinding costs from $172,000/year (1977-1978) to $40,000/year (1979-1980), for a $132,000 savings on that item alone. Each new or rebuilt motor delivered underwent a vibration acceptance test, including rolling element bearing analysis. A surprising 14% of new motors were rejected mostly for such bearing related problems as improper assembly or clearances, too tight a press fit, etc. They reported: "If we had put these motors in the field, they would have failed within 3 to 6 months." By finding a failing bearing or a coupling beginning to bind, overheating and machine failure was prevented, thus eliminating the need for rewinding. Vibration crews said: "The key result is increased lead time so we can properly schedule and have the right parts and people available on a no-panic basis." Before the program, operators wrote orders only when machines failed. Under the vibration control program, they now call for machine analysis whenever a change occurs, usually long before failure. If correction is not immediately possible, life prediction enables proper scheduling for shutdown and repair.

During that same period, the plant was expanded and the company increased its number of rotating machines by 41.8%. Ordinarily, such expansion would have required 20 more maintenance personnel. However, because of diligent vibration control, the number of millwright/mechanics was actually decreased from 63 in 1977 to 43 in 1980. Consider this in the light of a down economy which may require maintaining the same amount of machinery with a smaller crew.

This could give you the idea that we're in favor of putting people out of work. It is very far from the truth. Our conviction is that where plants are less efficient, less people have jobs. More productive plants almost always result in the kind of growth and healthy working conditions that produce more and more jobs, as well as profits. Throughout the world, the less efficient industries are constantly being replaced by the more efficient ones. Our company favors the worker by helping their companies eliminate waste - the kind of waste that drains away jobs as well as profits.

SPECIFIC EFFECTS ON MAINTENANCE COSTS IN DOLLARS PER HORSEPOWER

A medium sized oil refinery in Mississippi published a detailed financial report in the Oil & Gas Journal of October 18, 1976, documenting their maintenance cost reductions through their 5 year old vibration control program. This company had a unique way of measuring returns in "maintenance costs per horsepower." For 100 major machines, totaling 165,000 hp, annual costs were reduced from $7 per hp to $5 per hp. About 3900 "minor" machines (1,000 hp on down) such as ordinary pumps, motors, fans, etc., showed reduction of approximately 30% from $6.35 to $4.47 per hp per year. These "minor" machines totaled 340,000 hp. Simple arithmetic shows maintenance savings alone of almost $900,000 per year. Yet these maintenance savings were undoubtedly small compared to the extra production income.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

PRACTICAL INDEPTH VIBRATION TRAINING

The best equipment in the world is almost useless if those who use it have no practical, indepth training. Vibration training has to extend considerably beyond the instruments themselves and get into the "nuts and bolts" of just what is required to prevent, predict and correct almost any vibration problem that may arise. This cannot be done with a day of fast training by the local salesman. A practical "nuts and bolts" approach allows any person with a reasonable understanding of his plant's machinery and a fairly good reasoning ability, to readily learn all that he needs to know to carry out all phases of vibration analysis, prevention, and correction.

INTERPLAY BETWEEN SUPERVISORS, ENGINEERS, FOREMEN, CRAFTSMEN, AND TECHNICIANS

Another resistance to getting the proper financial returns is when all vibration control is only at a higher technical or engineering level, and not on the practical level of those closest to the machinery itself. Even though not everyone on the machinery team is full time in vibration control, vibration knowledge is important for:

Maintenance Engineers	*Area Supervisors*
Inspectors	*Rotating Equipment Specialists*
Reliability Engineers	*Shop Supervisors*
Advanced Mechanics	*Maintenance and Engineering Supervisors/Managers*

Too often Maintenance and Engineering Managers remain aloof from trying to understand vibration as they feel it is too complicated or technical. They get confused by the technical jargon used. Vibration knowledge can be brought down to the ordinary and practical level, and no supervisor who is responsible for large amounts of very expensive rotating machinery could really ignore or "delegate" the subject without having it show up in his work. A Maintenance Manager who attended one of Update's courses put it this way, "At first, I was sending only technical people who report to me to be trained, and found that at important meetings, they knew more about it than I did. Since I was responsible for machinery decisions that involved many thousands of dollars, sometimes for only one machine, I decided to get the training myself. Now I have a more complete perspective."

Then there are those plants or mills that have engineers or technicians doing a low level of vibration work that would soon bore a self-respecting engineer or technically trained person. For example, in the beginning, an engineer may enjoy field balancing. It is like a new toy to him. But after a few routine balancing jobs, he would subconsciously avoid working around the machinery, having " too many other things to do." This leads to actually working on only those situations that are extreme, ignoring those machines that would still run, even if "a little rough." The price is finally paid when the machines wear out bearings or seals that much sooner. The same applies for the routine analysis and procedures that could easily be performed by those that do not have as much education. Upon investigation, a fair percentage of vibration control can be preformed by people without formal higher education. Such people find this work a challenge, bringing their regular work out of the ordinary, less boring, and more importantly, get it done.

© Copyright 2001 Technical Associates Of Charlotte, P.C.

ACTUAL CASE HISTORIES OF VIBRATION DIAGNOSTICS ON VARIOUS MACHINE TYPES

SECTION	SUBJECT	PAGE
I. UNBALANCE		
A.	Unbalance Detection & Correction of a Belt-Driven Blower	2
B.	Unbalance Detection & Correction on an Air Conditioner Supply Fan	14
II. MISALIGNMENT		
A.	Effect of Coupling Misalignment on Vibration of a Water Scrubber Pump	18
B.	Detection & Correction of Sheave Misalignment	26
C.	Improvement of Vibration Following Alignment and Balancing of a Spencer Vacuum Pump	27
III. MECHANICAL LOOSENESS		
A.	Correction of Incinerator Fan with Loose Base and Unbalance	30
B.	In-Line Draft Inducer Fan Imbalance and Looseness	37
C.	Detection of Fan Wheel Rotor Looseness on a Nuva Feeder Vent Fan	40
IV. ROLLING ELEMENT BEARING PROBLEMS		
A.	Detection of Serious Wear of a Centrifugal Compressor Bullgear SKF 6213 Ball Bearing	44
B.	Severe Vibration Trend of a Timken Roller Bearing Installed in a Circulating Fan	55
C.	Detection of Serious Bearing Problems within a 400 HP Motor Serving a Hoffman Blower	61
D.	Surprising Detection of a Rolling Element Bearing Problem on a Draw Frame Input Pinion Bearing After Replacement of a Falk Gearbox Which Had Been Thought to be the Source of the Problem	67
E.	Serious Problem of a Bearing Turning on a Shaft of One Motor and Detection of Severe Bearing Problems on Another Motor as Evidenced by Numerous Outer Race Sum and Difference Frequencies	82
F.	Diagnosis of Problems with an Exhaust Fan Outboard Bearing Whose Manufacturer and Model Number were Unidentified	97
V. BELT DRIVE PROBLEMS		
A.	Significant Improvement in Motor and Blower Vibration Due to Belt Replacement	103
B.	Effect of Tightening Worn Belts on Motor Vibration	107
C.	Excessive Machine Tool Spindle Vibration Due to Belt Flutter	111

© Copyright 2001 Technical Associates of Charlotte, P.C.

DETECTION AND CORRECTION OF FAULTS ON AN AIR HANDLER HAVING PROBLEMS DUE TO UNBALANCE, SHEAVE MISALIGNMENT, DEFECTIVE BELTS, RESONANCE AND A CRACKED FRAME SUPPORTING THE OUTBOARD FAN PILLOWBLOCK BEARING

I. BACKGROUND

The assistance of Technical Associates was requested to help resolve continuing failure problems on the belt driven air handler driven by a 200 HP motor pictured in Figure 1. Here, the motor operated at a speed of 1185 RPM while the fan speed was approximately 427 RPM. For several years, this fan had experienced failures, the most dramatic of which had been three instances where the fan shaft actually broke in half. In addition, both the motor and fan had experienced continuing bearing and belt failure problems.

Upon arrival at the plant site, Technical Associates inspected some of the failed components of this machine including two of the broken shafts. Close inspection of the failed shafts did not show a distinct fatigue failure pattern either due to bending or shear. However, the races particularly of the fan bearings tended to indicate misalignment with most of the loading concentrated on one side of the race. Following below will be a summary of the results found during the analysis, recommendations made and a look at the improvement in vibration spectra before and after corrective actions were taken.

II. RESULTS AND CONCLUSIONS

Vibration measurements were taken on each of the motor and fan bearing housings pictured in Figure 1. Plot 1 taken on the outboard motor bearing in the axial direction (Position 1A in Figure 1) showed high axial vibration with an overall level of .632 in/sec. Surprisingly, the motor running speed vibration at about 1192 CPM was only .090 in/sec. However, the dominant vibration on the motor was at 427 CPM with an excessive magnitude of .585 in/sec. Of course, the source of the 427 CPM vibration was the operating speed of the fan. Thus, this was a classic case often indicating sheave misalignment which will often generate excessive vibration at fan running speed in the axial direction when making measurements on the motor.

Next, another surprising spectrum is shown in Plot 2 which was taken in the axial direction on the sheave-side fan bearing at Position 3A (see Figure 1).

Here, the vibration at 427 fan speed was low, but there was a very pronounced frequency at only 300 CPM dominating the vibration. Since this was suspected to be a frequency generated by belt speed (or multiple), the client was requested to review the purchase order for the motor and fan sheaves to confirm their pitch diameters as well as confirming the length of

© Copyright 2001 Technical Associates of Charlotte, P.C.

the belts. After this information was given, the belt speed (RPM) was calculated to be about 149.5 RPM (see Note 1). Therefore, this revealed the source of the 300 CPM vibration to be 2X belt defect frequency. It is not unusual for defects on belts to first generate a frequency at 2X belt speed (since the belt contacts two sheaves per revolution). However, what was unusual was the significant amplitude that resulted here at Position 3A in Plot 2. Therefore, the client was requested to stop the air handler to allow performance of impulse natural frequency tests. Such natural frequency testing was performed on each of the motor and fan bearing housings in horizontal, vertical and axial directions. Not surprisingly, when the test was performed on the sheave-side bearing in the axial direction, a pronounced axial natural frequency was found at 315 CPM as shown in Plot 3 (or only 5% away from the 300 CPM peak caused by 2X belt defect frequency). Therefore, even when defects would occur in the belts, their vibration would be greatly amplified by as much as 10 to 30 times higher due to resonance. And, if the belts suffered significant wear, resultant axial amplitudes would likely increase to highly unaceptable levels.

Next, the spectrum shown in Plot 4 indicated unbalance of the fan wheel as evidenced by a high amplitude .427 in/sec vibration at 427 RPM fan speed with the data taken on the same sheave-side bearing, but at Position 3H in the horizontal direction. Interestingly, note that none of the vibration at 2X belt defect frequency appeared whatsoever in the spectrum taken in this direction - another indication of the significant resonance problem this fan had in its axial direction. Phase analysis data was captured to ascertain the cause of the high 1X RPM fan vibration. This phase data is shown on Figure 1. Looking at the data taken on Positions 3 and 4, note that the phase difference in the horizontal direction between the outboard and inboard fan bearings (Postions 3 and 4) was approximately 110° while the vertical phase difference was approximately 130° between these two bearings. Since the phase difference was approximately the same in both the horizontal and vertical directions, and due to the high vibration 1X RPM, fan wheel unbalance was suspected.

NOTES:

1. $$\textit{Belt Speed} = \frac{\pi \times \textit{Pulley RPM} \times \textit{Pulley Pitch Diameter}}{\textit{Belt Length}}$$

   (If Fan Sheave Pitch Diameter is entered, then Fan Sheave RPM must be also; and vice versa for the motor sheave.)

III. RECOMMENDATIONS

A. It was recommended that the belts be replaced and a close inspection be made of both the motor and fan sheaves to confirm they were in good condition. After replacing the belts (and sheaves if necessary), it was recommended to precision align the motor and fan sheaves with respect to one another.

© Copyright 2001 Technical Associates of Charlotte, P.C.

B. It was also recommended that the frame of the fan be stiffened in the axial direction to raise its natural frequency above 315 CPM which would come very close to 2X belt speed when future belt defects did appear. However, it was cautioned that the natural frequency should not be raised to next coincide with either 3X or 4X belt speed. In other words, when stiffening attempts were made, an FFT analyzer was recommended to be on hand to ensure to what frequencies the frame natural frequency would be shifted.

C. After completion of recommendations A and B, it was recommended that a fan wheel balance should be performed in place to significantly lower the vibration at 1X fan speed below the .427 in/sec measured on May 18.

IV. RESULTS OF CORRECTIVE ACTIONS

Corrective actions were attempted by Technical Associates on the same test day (May 18) as well as after this test. First, a hydraulic jack was placed in the axial direction on the fan frame in an attempt to temporarily shift the axial natural frequency away from 2X belt speed (and below 3X belt speed). However, the frame itself was of insufficient cross section to provide the necessary restraint to noticeably shift the natural frequency. Therefore, it was recommended on May 18 that the center distance between the motor and fan sheaves be changed which would not shift the frame stiffness or natural frequency, but would move the belt speed away from coincidence with the axial natural frequency.

Followup tests were performed on June 22 after the belts had been replaced and the motor had been moved further away from the fan, thereby increasing the center distance. This increase in center distance lowered the belt speed from about 150 RPM down to approximately 125 RPM. Therefore, now neither 2X belt speed (250 CPM) nor 3X belt speed (375 CPM) would be close to the fan frame axial resonance of about 315 CPM. Next, a fan wheel balance was performed in-place by Technical Associates on June 22. Plots 5 and 6 show the results of this balancing exercise on the inboard (Pos. 3H) and outboard (Pos. 4H) fan bearing. Even though the fan wheel showed noticeable wobble when it was slow-rolled, Plot 5 taken on Position 3H showed the field balance lowered vibration at fan speed from .444 in/sec down to .052 in/sec; while Plot 6 taken on Position 4H showed that it lowered the 1X RPM level from .365 in/sec down to only .052 in/sec.

However, please note the peak at 2X fan running speed in Plot 6 taken on Position 4H on the outboard fan bearing. Note that its level did not decrease (in fact, slightly increased to .241 in/sec) after the balance exercise. An examination of the frame supporting the outboard fan pillowblock bearing revealed a crack in the frame itself. This was pointed out to the client personnel who were advised to either replace and/or weld this frame.

© Copyright 2001 Technical Associates of Charlotte, P.C.

V. LATER FOLLOWUP AFTER WELDING OF THE OUTBOARD FAN PILLOWBLOCK FRAME

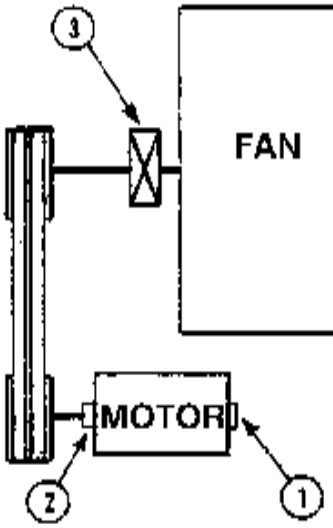
Plot 7 shows the spectrum taken after the frame supporting the outboard pillowblock fan bearing was welded. It lowered the vibration at 2X RPM from .241 in/sec down to .078 in/sec. Note that this spectrum was taken some time after the original corrective actions and that the vibration at 1X fan speed had now increased somewhat up to .193 in/sec. A check with the client showed that this was likely to buildup on the fan wheel. Still, a level less than .20 in/sec at 1X RPM on this fan wheel was quite acceptable and the noticeable drop in 2X RPM vibration was beneficial.

© Copyright 2001 Technical Associates of Charlotte, P.C.

Technical Associates of Charlotte (704-333-9911)

MACHINERY DATA GUIDE SHEET

RELIABILITY SURVEY — predictive maintenance program — (Page__of__)



MOTOR SPEED PHASE ANALYSIS

POS.	DIRECT PHASE	CORRECTED PHASE
1A	140°	140°
1H	210°	210°
1V	140°	140°
2A	305°	125°
2H	170°	170°
2V	220°	220°

FAN SPEED PHASE ANALYSIS

POS.	DIRECT PHASE	CORRECTED PHASE
3A	260°	260°
3H	300°	300°
3V	340°	340°
4A	100°	280°
4H	190°	190°
4V	250°	250°

ΔHORIZ 3-4 = 300° - 190° = 110°
ΔVERT 3-4 = 340° - 250° = 90°

AREA:_____

MACHINE IDENT: L-12 AIR HANDLER ROUTE:_____

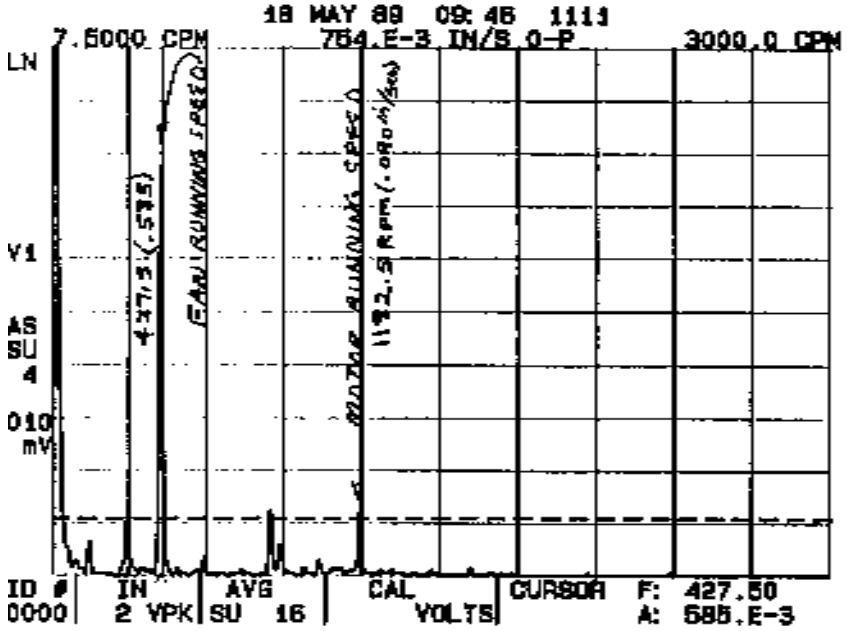
DRIVER	INTERMEDIATE	DRIVEN
RPM min : 1185	RPM in :	RPM min : 427.5
max :	out :	max :
MFGR :	RATIO:	MFGR. :
MODEL :	MODEL :	MODEL :
SER NO. :	SER NO. :	SER NO. :
V___ P___ A___	RATING :	RATING :
SHEAVE : ___DIA.	BELT # :	#BLADES :
BRG#: ___(I)	BRG#: ___(I)	BRG#: NU322 (I)
BRG#: ___(O)	BRG# ___(O)	BRG#: 6320 (O)
H.P.: 200	GEAR T: ___CT.	GEAR T: ___CT.
OTHER:	OTHER:	OTHER:

Π-112492-1

FIGURE 1

© Copyright 2001 Technical Associates of Charlotte, P.C.

© Copyright 2001 Technical Associates of Charlotte, P.C.



IN/S 0--P	CPM
585.E-3	427.50
92.9E-3	847.50
89.5E-3	1192.5

PEAK PICK ON TRACE 1

OVERALL = .632 in/sec

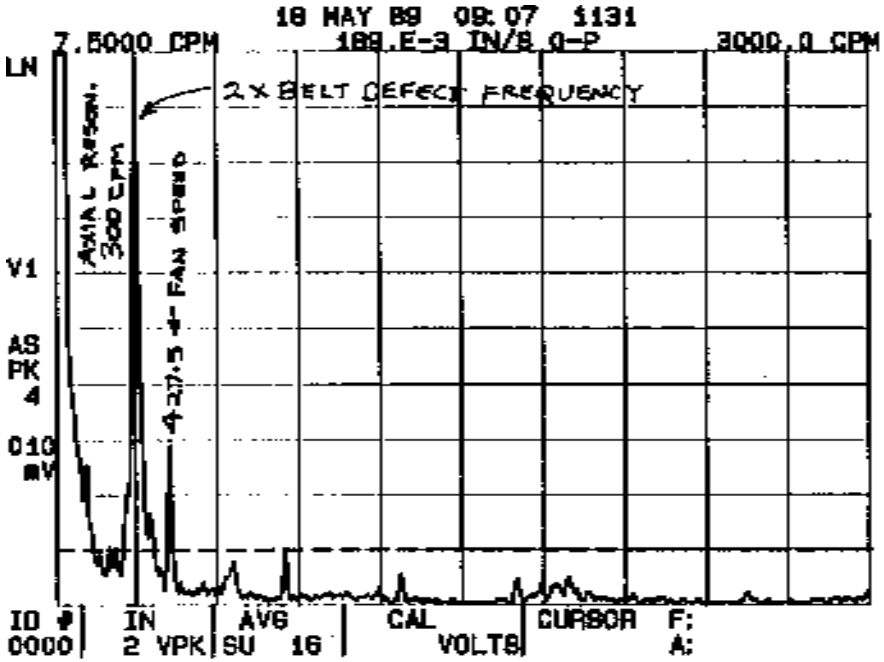
24-MAY-89

13: 17: 04

MOTOR VIBRATION
AXIALLY ON OUTBOARD END
POSITION 1A

PLOT 1

L-12AH



PEAK PICK ON TRACE 1

IN/S 0-P	CPM
48.3E-3	112.50
19.8E-3	202.49
19.8E-3	225.00
38.8E-3	270.00
171.E-3	300.00
31.9E-3	352.50
55.2E-3	427.50

24-MAY-89

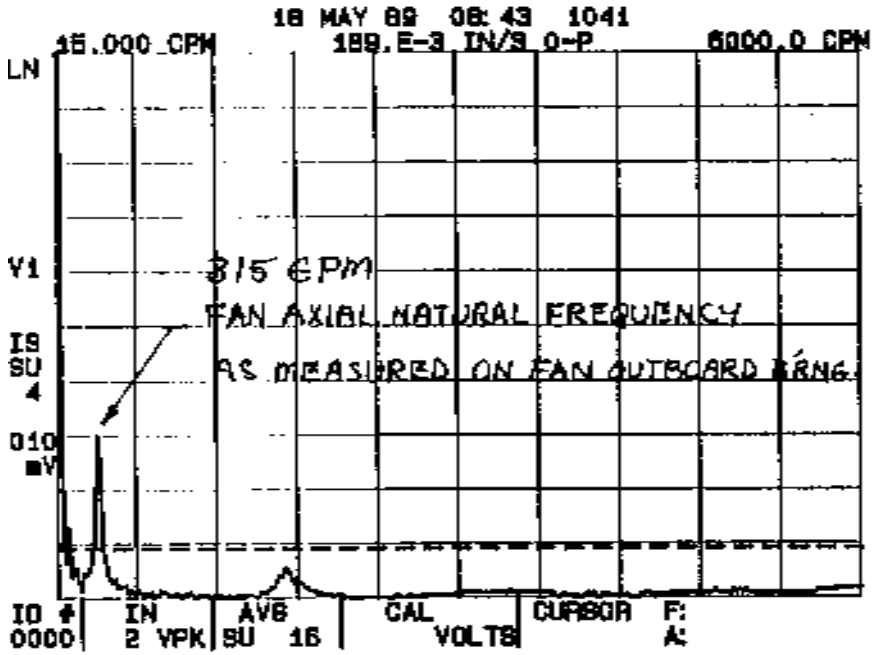
12: 52: 05

INBOARD FAN BEARING
AXIAL - PEAK HOLD
POSITION 3A
PLOT 2

L-12AH

© Copyright 2001 Technical Associates of Charlotte, P.C.

© Copyright 2001 Technical Associates of Charlotte, P.C.



IN/S 0-P	CPM
25.0E-3	90.000
57.0E-3	315.00

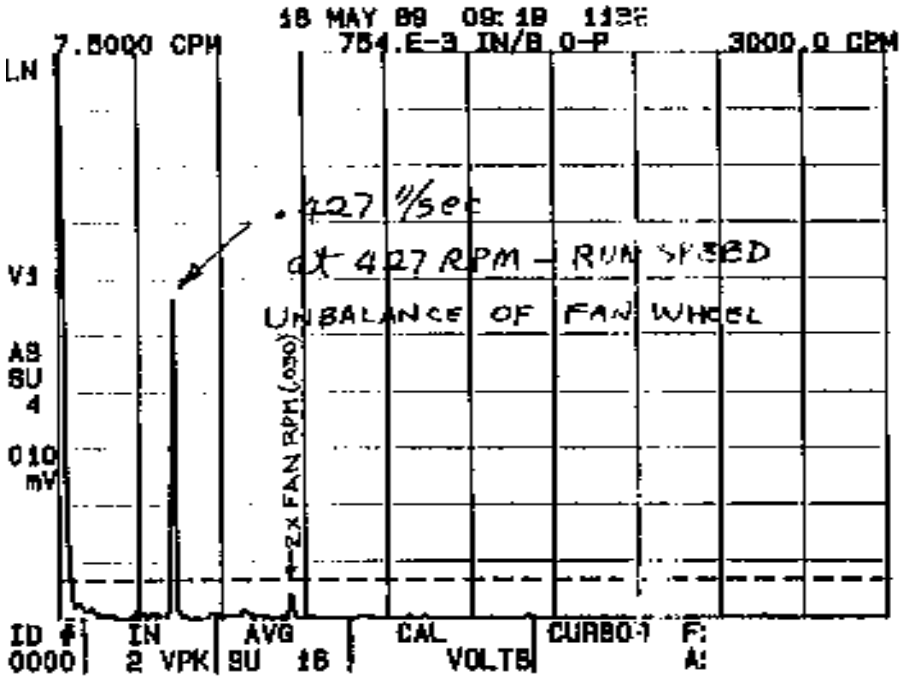
PEAK PICK ON TRACE 1

24-MAY-89

12: 36: 05

IMPULSE TEST ON FAN OUTBOARD BEARING
POSITION 3A - IMPULSE IN AXIAL DIRECTION
PLOT 3

L-12AH



PEAK PICK ON TRACE 1

IN/S 0-P	CPM
427.E-3	427.50

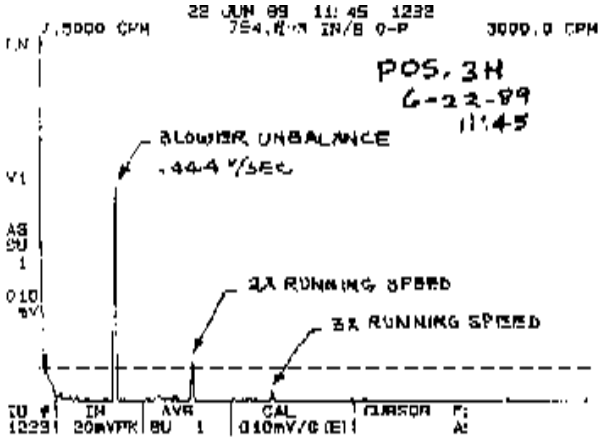
24-MAY-89

12: 55: 56

INBOARD FAN BEARING - HORIZONTAL
POSITION 3H
PLOT 4

L-12 AH

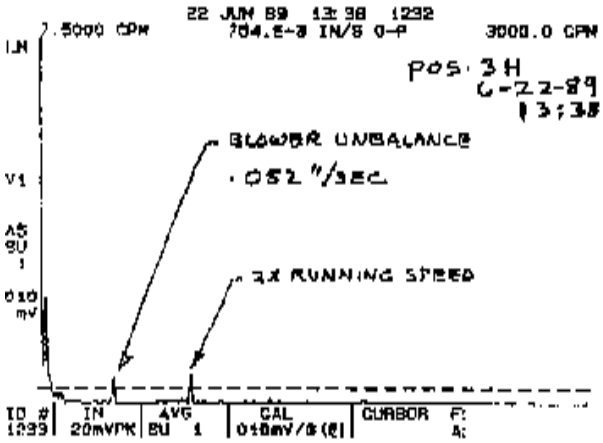
© Copyright 2001 Technical Associates of Charlotte, P.C.



PEAK PICK ON TRACE 1

IN/S 0-P	CPM
99.8E-3	45.000
444.E-3	427.50
82.6E-3	855.00

BEFORE BALANCING



PEAK PICK ON TRACE 1

IN/S 0-P	CPM
217.E-3	52.500
51.6E-3	427.50
58.5E-3	855.00

AFTER BALANCING

22-JUN-89

15: 34: 00

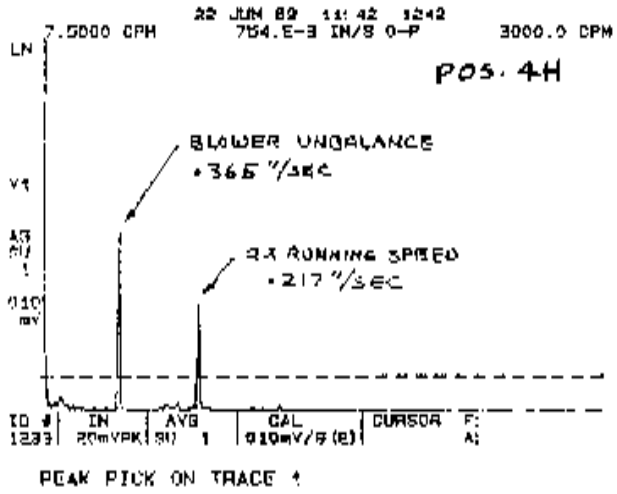
UPPER PLOT = BEFORE BALANCING
.444"/sec

L-12 FAN

LOWER PLOT = AFTER BALANCING
.052"/sec

PLOT 5

© Copyright 2001 Technical Associates of Charlotte, P.C.

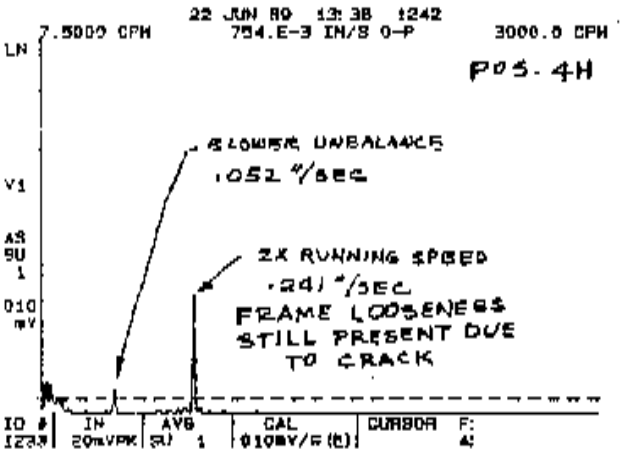


PEAK PICK ON TRACE 1

10

IN/S 0-P	CPM
365.E-3	427.50
217.E-3	855.00

BEFORE BALANCING



PEAK PICK ON TRACE 1

IN/S 0-P	CPM
65.4E-3	60.000
62.0E-3	82.500
51.6E-3	427.50
241.E-3	855.00

AFTER BALANCING

22-JUN-89

15: 31: 17

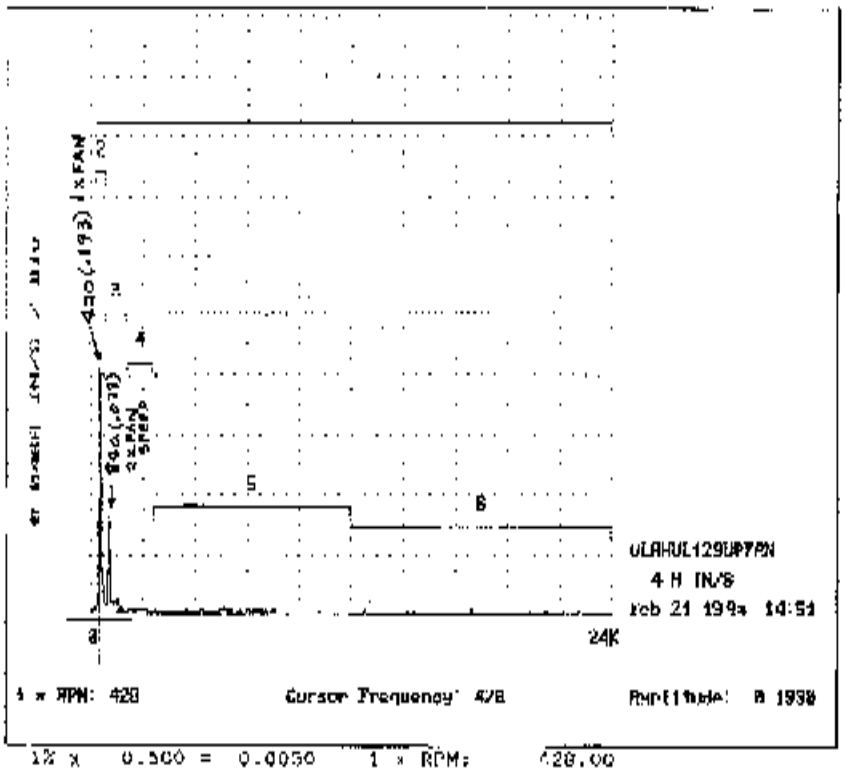
UPPER PLOT = BEFORE BALANCING
.365"/sec

L-12 FAN

LOWER PLOT = AFTER BALANCING
.052"/sec

PLOT 6

© Copyright 2001 Technical Associates of Charlotte, P.C.



Machine	Pos	Dir	Units	Date
ULAHUL12S:IPFAN	4	H	IN/S	Feb 21 1960 14:51

Frequency	1 x RPM	Amplitude
420	0.98	0.1930
840	1.96	0.0775
1260	2.94	0.0125
6130	14.44	0.0060

AFTER WELDING FRAME SUPPORTING OUTBOARD FAN PILLOWBLOCK BEARING

PLOT 7

T - Threshold Adjust Q - Quit

© Copyright 2001 Technical Associates of Charlotte, P.C.

DETECTION & CORRECTION OF UNBALANCE ON AN AIR CONDITIONER SUPPLY FAN

I. BACKGROUND

An HVAC contractor had been having vibration and associated noise difficulties with one of their customer's HVAC circulating fans. The contractor had changed bearings, adjusted belts, etc., without any success in improving vibration or noise levels and requested that Technical Associates perform a diagnostic investigation to determine the problem source.

II. RESULTS

The unit was belt driven as shown in Figure 1 with one (1) center mounted squirrel cage fan and one (1) overhung squirrel cage fan. The motor rotated at 1738 RPM and the fan at 738 RPM. All the spectra collected were dominated by a 720 CPM vibration with the highest amplitudes (axial and horizontal) at Position 4 on the fan shaft. It was determined that fan imbalance was the problem, and since Position 4 had the highest amplitude, balancing the overhung fan would produce the best results. What followed was a successful single plane balance with measurements from the Position 4 bearing. The weights were added at the inboard end of the overhung fan wheel on its backing plate close to bearing 4. Vibration levels at Position 4 fell from 1.74 in/sec at 720 CPM (fan rotational speed) to .035 in/sec at 720 CPM after the balance as seen by the "Before" and "After" spectra of Figure 2. The axial vibration at Position 4 also improved from 1.21 in/sec "Before" the balance to .448 in/sec "After" as seen in the Figure 3 spectra. The remaining axial response was attributed to:

1. Inadequate axial support
2. The variable pitch motor sheave
3. Sheave misalignment.

Noise levels also fell with the improved vibration.

III. CONCLUSIONS

1. The dominant problem was fan imbalance (168 grams of correction weights added to the outboard fan reduced unbalance to acceptable levels).
2. The axial response that remained after the balance was likely a result of:
   a. Structural flexing at the bearing supports
   b. The variable pitch motor sheave
   c. Sheave/belt misalignment.

© Copyright 2001 Technical Associates of Charlotte, P.C.

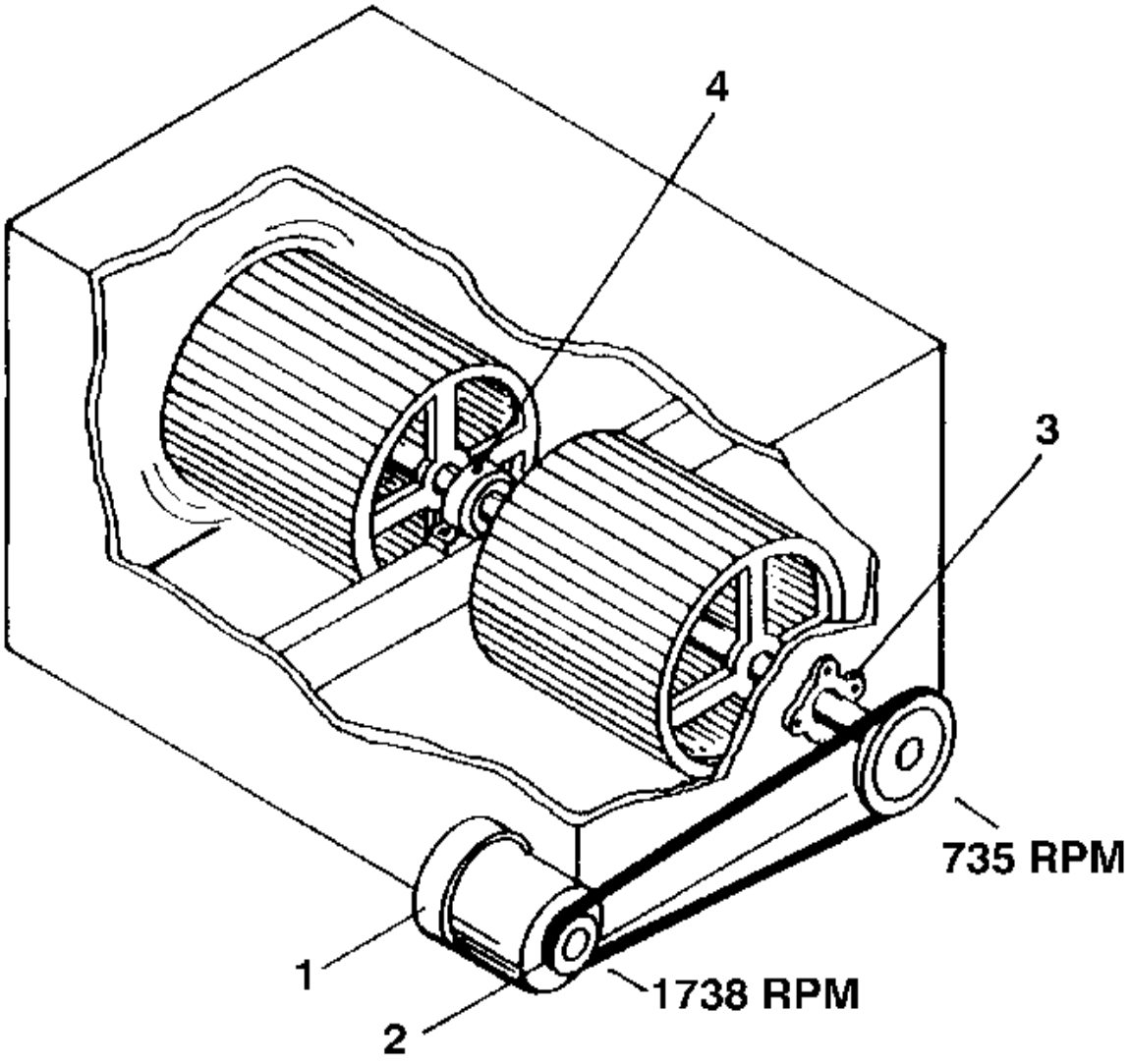
3. Noise levels improved along with the vibration.

4. The simple single plane balance procedure worked well in correcting a significant imbalance in a complicated fan arrangement.

IV. RECOMMENDATIONS

1. Replace the motor's variable pitch sheave with a fixed diameter sheave of the proper size.

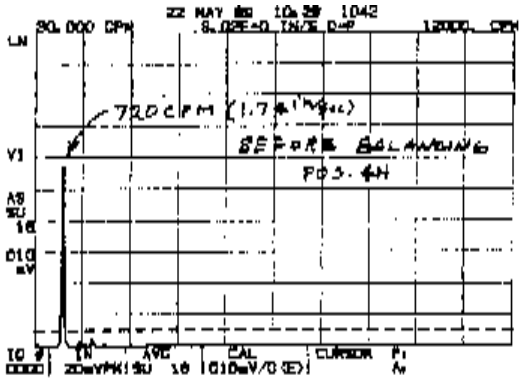
2. Align and tension the belts to specification.



SQUIRREL CAGE FAN ARRANGEMENT
FIGURE 1

CF10 R-9704-1

© Copyright 2001 Technical Associates of Charlotte, P.C.



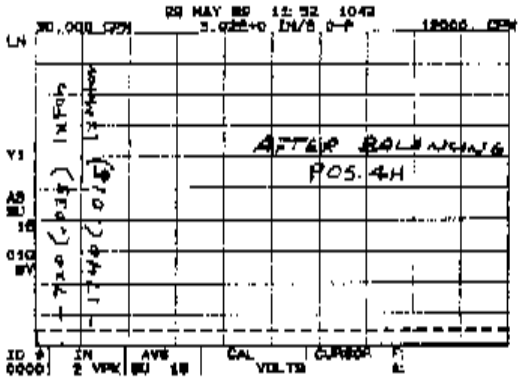
IN/S 0-P	CPM
1.74	720.00

PEAK PICK ON TRACE 1

OA – 2.03
POSITION 4H

23-MAY-89

16: 33: 40



IN/S 0-P	CPM

PEAK PICK ON TRACE 1

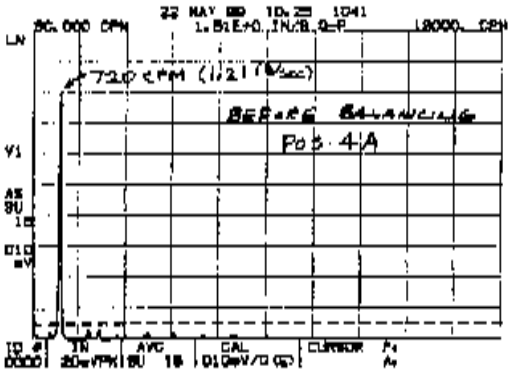
PRECISION BALANCED

24-MAY-89

09: 26: 19

POSITION 4
HORIZONTAL "BEFORE" AND "AFTER" BALANCE SPECTRA
FIGURE 2

© Copyright 2001 Technical Associates of Charlotte, P.C.

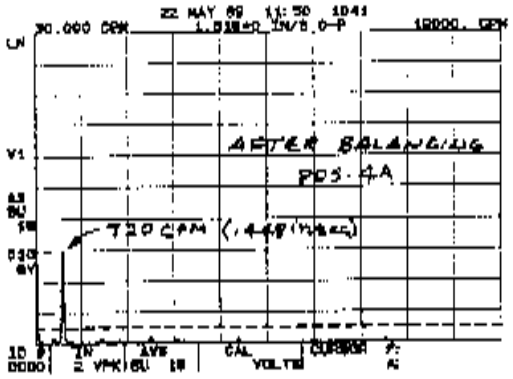


IN/S 0-P	CPM
1.21	720.00

OA = 1.42
POSITION 4A

23-MAY-89

16:32:14



IN/S 0-P	CPM
448.E-3	720.00

1 INADEQUATE AXIAL SUPPORT BY FRAME (Esp. at IB Fan Brg.)
2 VARIABLE PITCH MOTOR SHEAVE IS PROBLEM
3 SHEAVES HAVE SOME MISALIGNMENT

24-MAY-89

RECOMMEND FIXED SHEAVE

09:25:34

POSITION 4
AXIAL "BEFORE" AND "AFTER" BALANCE SPECTRA
FIGURE 3

© Copyright 2001 Technical Associates of Charlotte, P.C.

EFFECT OF COUPLING MISALIGNMENT ON VIBRATION OF A WATER SCRUBBER PUMP

I. BACKGROUND

During a July 13, 1989 predictive maintenance vibration survey by Technical Associates, a horizontally mounted centrifugal water pump (1-1 Scrubber Pump) was detected as having a sudden axial vibration increase. See Figure 1 for the machine arrangement. The increase is shown in the trend graphs and waterfall plots of Figures 2, 3 and 4 taken on Positions 1A and 2A as shown. Notice that the waterfall spectra show the dominant vibration was at 1X RPM (nominal 1800 RPM). Since the motor was having other problems, our client decided to replace the motor instead of just aligning it. A new coupling was unavailable at that time, so the old coupling was reused. The next survey of 9/7/89 had reduced axial measurements indicating the unit was well aligned as seen by the trends and waterfall plots of Figures 2 and 4. The axial vibration remained stable until the October 5, 1989 survey. This survey found the axial motor vibration had again increased to levels in excess of 1.0 in/sec as seen in Figures 2, 3 and 4. This time, the problem was found to be caused by excessive wear of the "used" coupling. After the coupling was replaced and the unit aligned, vibration levels again fell to acceptable levels as seen in the 11/9/89 measurements of Figure 5 and the trends and spectra of Figures 2 and 3.

II. CONCLUSIONS

A. 1-1 Scrubber Pump's first misalignment condition resulted from a motor problem and the second from a coupling problem.

B. Vibration measurements after the final alignment indicate the unit was well aligned.

III. RECOMMENDATIONS

A. Recommend continuing the periodic vibration measurements on a monthly basis.

IV. RESULTS OF CORRECTIVE ACTIONS

A. The final alignment with the new coupling has remained stable, at least through November, 1991 as seen in the Trend Graphs of Figures 6 and 7.

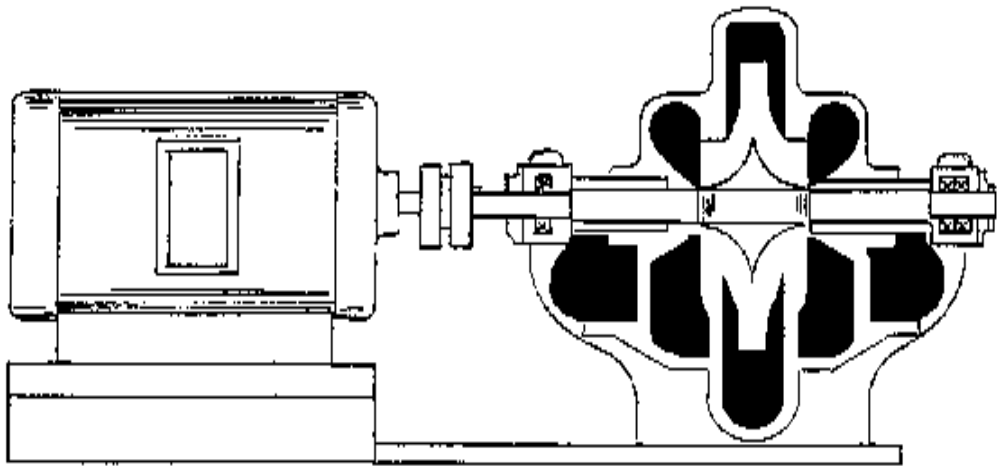
HP1 R-9208-1

© Copyright 2001 Technical Associates of Charlotte, P.C.

Technical Associates of Charlotte (704-333-9011)

MACHINERY DATA GUIDE SHEET

RELIABILITY SURVEY predictive maintenance program (Page__of__)



MACHINE IDENT: 1-1 SCRUBBER PUMP

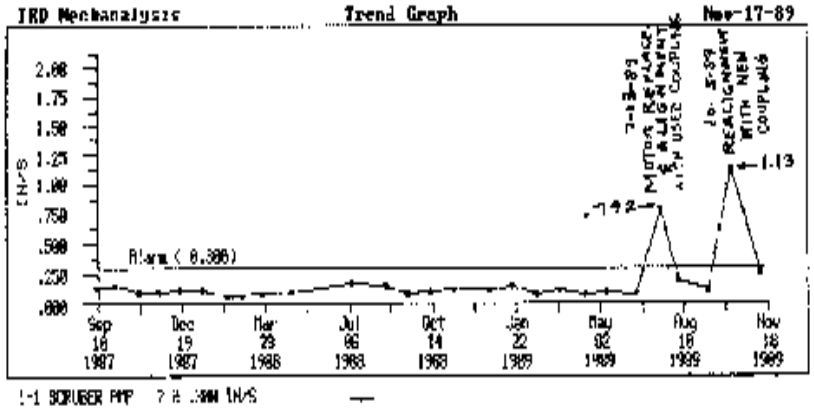
AREA:

ROUTE:

DRIVER	INTERMEDIATE	DRIVEN
RPM min :	RPM in :	RPM min :
max : 1770	out :	max :
MFGR : TOSHIBA	RATIO:	MFGR. : WORTHINGTON
MODEL : B2004FKF4BD	MODEL :	MODEL : 10LR15
SER NO. : 4903316	SER NO. :	SER NO. : 60-740830
V 460 P 3 A 230	RATING :	RATING :
SHEAVE : DIA.	BELT # :	#BLADES :
BRG#: (I)	BRG#: (I)	BRG#: (I)
BRG#: (O)	BRG# (O)	BRG#: (O)
H.P.: 200	GEAR T: CT.	GEAR T: CT.
OTHER:	OTHER:	OTHER:

FIGURE 1

© Copyright 2001 Technical Associates of Charlotte, P.C.



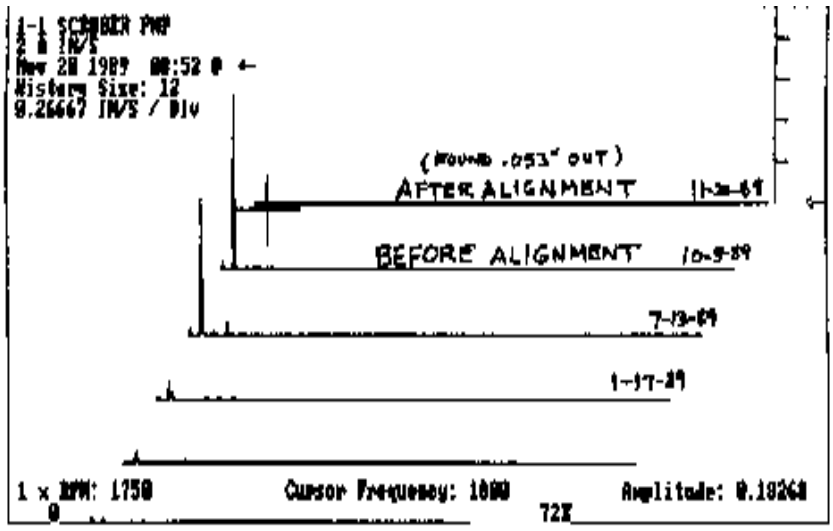
IRD Mechanalysis Data Analysis System

Machine 1-1 SCRUBBER FAN Pos V Dir A Units IN/S Alarm .300

Ampl	Insp Code	Date mm/dd/yy	Time	Ampl	Insp Code	Date mm/dd/yy	Time
.134	0	09-10-87	0:00	.114	0	11-07-88	12:16
.159	0	09-30-87	0:00	.122	0	12-20-88	12:46
.099	0	10-28-87	0:00	.147	0	01-17-89	11:05
.099	0	11-20-87	0:00	.075	0	02-14-89	10:47
.114	0	12-16-87	0:00	.117	0	03-14-89	10:44
.114	0	01-11-88	0:00	.078	0	04-13-89	12:00
.066	0	02-11-88	0:00	.104	0	05-09-89	9:23
.004	0	03-23-88	0:00	.078	0	06-13-89	12:57
.094	0	04-27-88	0:00	.742	0	07-13-89	11:34
.164	0	07-09-88	0:00	.170	0	08-03-89	13:10
.155	0	08-15-88	12:43	.116	0	09-07-89	10:49
.085	0	09-13-88	15:12	1.13	0	10-05-89	10:51
.099	0	10-11-88	12:46	.258	0	11-09-89	12:20

FIGURE 2

© Copyright 2001 Technical Associates of Charlotte, P.C.



2% x 2.0000 = 0.04000 1 x RPM: 1750.00

Machine	Pos	Dir	Units	Date
1-1 SCRUBER PMP	2	A	IN/S	Nov 20 1989 08:52

Frequency	1 x RPM	Amplitude
1800	1.03	0.18268

2% x 2.0000 = 0.04000 1 x RPM: 1750.00

Machine	Pos	Dir	Units	Date
1-1 SCRUBER PMP	2	A	IN/S	Oct 05 1989 10:51

Frequency	1 x RPM	Amplitude
180	0.10	0.04988
1800	1.03	1.09992

2% x 2.0000 = 0.04000 1 x RPM: 1750.00

Machine	Pos	Dir	Units	Date
1-1 SCRUBER PMP	2	A	IN/S	Jul 13 1989 11:34

Frequency	1 x RPM	Amplitude
180	0.10	0.04988
1800	1.03	0.82986
5400	3.09	0.08483

2% x 2.0000 = 0.04000 1 x RPM: 1750.00

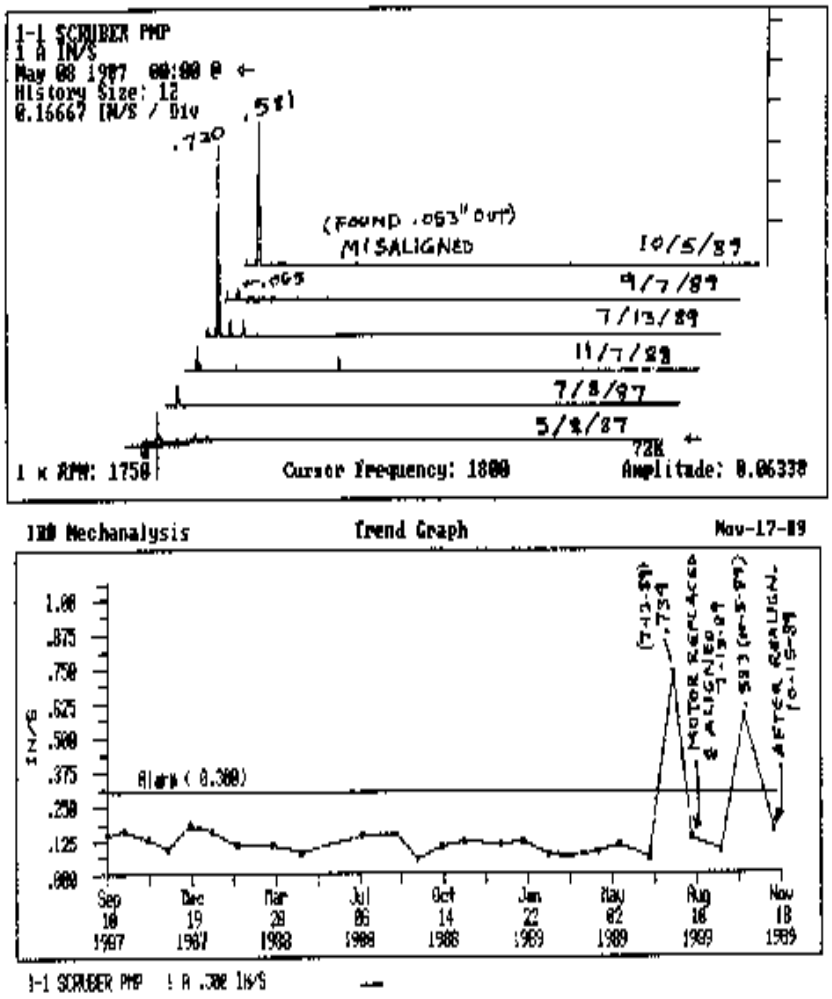
Machine	Pos	Dir	Units	Date
1-1 SCRUBER PMP	2	A	IN/S	Jan 17 1989 11:05

Frequency	1 x RPM	Amplitude
1800	1.03	0.10899

FIGURE 3

© Copyright 2001 Technical Associates of Charlotte, P.C.

EFFECT ON VIBRATION OF ALIGNMENT



IRD Mechanalysis Data Analysis System

Machine 1-1 SCRUBER PMP Pos 1 Dir A Units IN/S Alarm .300

Ampl	Insp Code	Date mm/dd/yy	Time
.144	0	09-10-87	0:00
.164	0	09-30-87	0:00
.129	0	10-30-87	0:00
.094	0	11-20-87	0:00
.184	0	12-16-87	0:00
.159	0	01-11-88	0:00
.109	0	02-11-88	0:00
.109	0	03-23-88	0:00
.079	0	04-27-88	0:00
.144	0	07-09-88	0:00
.145	0	08-18-88	12:42
.060	0	09-13-88	15:11
.099	0	10-11-88	12:45
.124	0	11-07-88	12:15
.111	0	12-20-88	12:45
.117	0	01-17-89	11:04
.077	0	02-14-89	10:47
.069	1	03-14-89	10:43
.083	0	04-13-89	12:00
.114	0	05-09-89	9:33
.061	0	06-13-89	12:57
.734	0	07-13-89	11:53
.140	0	08-03-89	13:09
.085	0	09-07-89	10:49
.583	0	10-05-89	10:49
.171	0	11-09-89	12:19

FIGURE 4

© Copyright 2001 Technical Associates of Charlotte, P.C.

IRD Mechanalysis Data Analysis System — Page 1

Nov-17-89

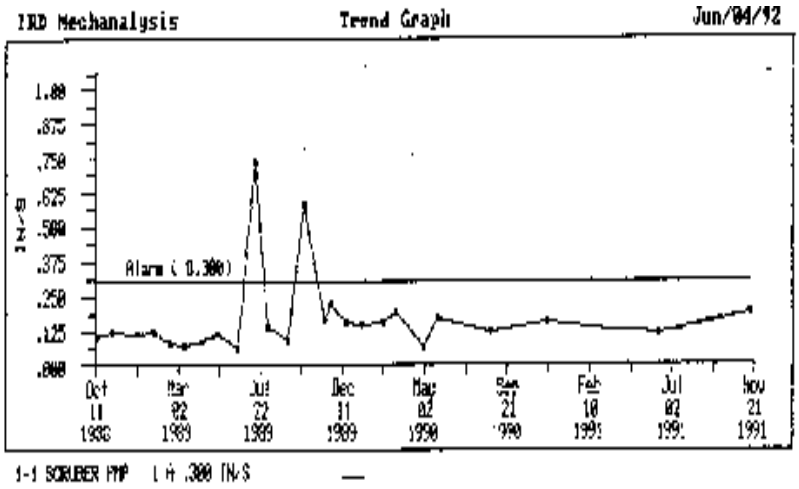
Machine Report (Latest Data) for: 1-1 SCRUBER FMP

Date	Time	Pos	Dir	Ampl	Units	Change	Alarm	Inspection
Nov-09-89	12:19	1	A	.171	IN/S	-71%	.300	
Nov-09-89	12:19	1	H	.079	IN/S	-64%	.300	
Nov-09-89	12:19	1	H	.262	g/SE	134%	.600	←
Nov-09-89	12:19	1	V	.071	IN/S	-49%	.300	Arrows Show Little Improvement
Nov-09-89	12:20	2	A	.258	IN/S	-77%	.300	in Spike Energy as Expected
Nov-09-89	12:20	2	H	.061	IN/S	-73%	.300	
Nov-09-89	12:20	2	H	.175	g/SE	-8%	.600	←
Nov-09-89	12:20	2	V	.174	IN/S	-61%	.300	
Nov-09-89	12:20	3	A	.202	IN/S	2%	.300	
Nov-09-89	12:20	3	H	.116	IN/S	10%	.300	
Nov-09-89	12:20	3	H	.228	g/SE	24%	.600	←
Nov-09-89	12:20	3	V	.156	IN/S	47%	.300	
Nov-09-89	12:21	4	A	.170	IN/S	-1%	.300	
Nov-09-89	12:21	4	H	.112	IN/S	-7%	.300	
Nov-09-89	12:21	4	H	.127	g/SE	7%	.600	←
Nov-09-89	12:21	4	V	.118	IN/S	-2%	.300	

16 lines this report

FIGURE 5

© Copyright 2001 Technical Associates of Charlotte, P.C.



IRD Mechanalysis Data Analysis System

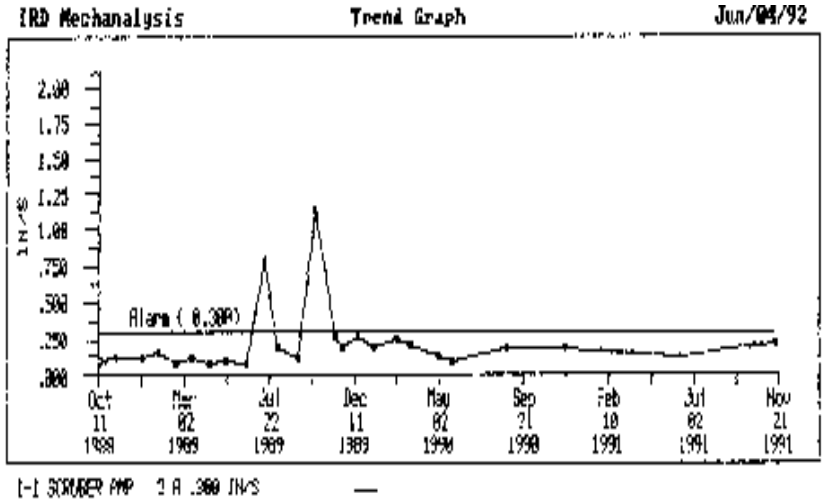
Machine 1-1 SCRUBER FMP Pos 1 Dir A Units IN/S Alarm .300

Ampl	Insp Code	Date mm/dd/yy	Time
.099	0	10/11/88	12:45
.124	0	11/07/88	12:15
.111	0	12/20/88	12:45
.117	0	01/17/89	11:04
.077	0	02/14/89	10:47
.068	1	03/14/89	10:43
.088	0	04/13/89	12:00
.111	0	05/09/89	9:33
.061	0	06/13/89	12:57
.734	0	07/13/89	11:33
.140	0	08/03/89	13:09
.085	0	09/07/89	10:47
.583	0	10/05/89	10:49

Ampl	Insp Code	Date mm/dd/yy	Time
.171	0	11/09/89	12:19
.213	0	11/20/89	8:50
.160	0	12/15/89	10:08
.151	0	01/11/90	9:01
.154	0	02/20/90	8:21
.194	0	03/13/90	8:17
.067	0	04/30/90	7:15
.168	0	05/24/90	10:36
.169	0	06/21/90	10:36
.135	0	08/23/90	11:14
.154	0	11/29/90	10:59
.110	0	03/04/91	10:31
.190	0	11/14/91	9:04

FIGURE 6

© Copyright 2001 Technical Associates of Charlotte, P.C.



IRD mechanalysis Data Analysis System

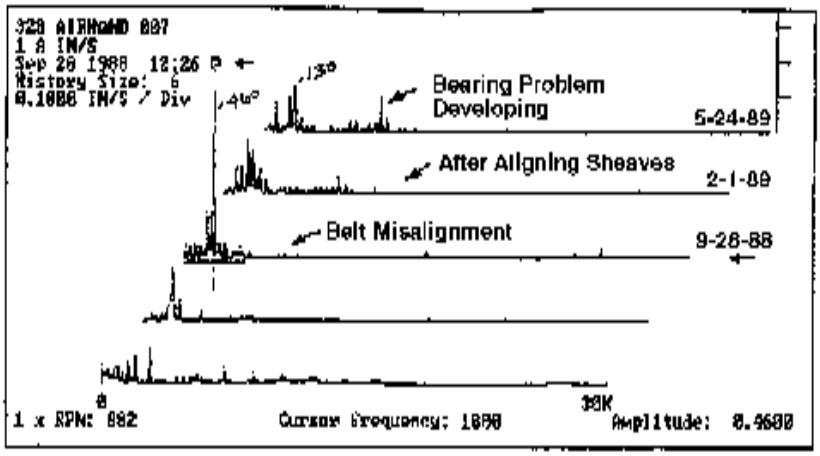
Machine 1-1 SCRUBER PMP Pos 2 Dir A Units IN/S Alarm .700

Ampl	Insp Code	Date mm/dd/yy	Time	Ampl	Insp Code	Date mm/dd/yy	Time
.099	0	10/11/88	12:46	.058	0	11/09/90	12:20
.114	0	11/07/88	12:16	.037	0	11/20/89	8:32
.122	0	12/20/88	12:46	.257	0	12/15/89	10:08
.147	0	01/17/89	11:00	.182	0	01/11/90	9:01
.075	0	02/14/89	10:47	.144	0	02/20/90	8:32
.117	0	03/14/89	10:44	.107	0	03/26/90	9:20
.078	0	04/12/89	12:00	.174	0	04/30/90	9:45
.103	0	05/09/89	9:53	.103	0	05/24/90	10:37
.079	0	06/13/89	12:57	.103	0	05/24/90	10:37
.772	0	07/13/89	11:34	.194	0	09/28/90	11:35
.150	0	08/03/89	13:10	.194	0	11/29/90	10:59
.116	0	09/07/89	10:49	.113	0	06/06/91	10:30
1.13	0	10/05/89	10:51	.228	0	11/14/91	9:04

FIGURE 7

© Copyright 2001 Technical Associates of Charlotte, P.C.

DETECTION & CORRECTION OF SHEAVE MISALIGNMENT



5% x 1.000 = 0.0500 1 x RPM: 882.00

Machine	Pos	Dir	Units	Date
328 AIRHAND 007	1	A	IN/S	May 24 1989 12:16

Frequency	1 x RPM	Amplitude
750	0.85	0.0824
1425	1.62	0.0975
1800	2.04	0.1399
6825	7.74	0.1025

5% x 1.000 = 0.0500 1 x RPM: 882.00

Machine	Pos	Dir	Units	Date
328 AIRHAND 007	1	A	IN/S	Feb 01 1989 13:06

Frequency	1 x RPM	Amplitude
750	0.85	0.0750
1050	1.19	0.0824
1425	1.62	0.1525
1725	1.96	0.1174
2175	2.47	0.0699

0025% x 1.000 = 0.0300 1 x RPM: 882.00

Machine	Pos	Dir	Units	Date
328 AIRHAND 007	1	A	IN/S	Sep 28 1988 12:26

Frequency	1 x RPM	Amplitude
75	0.09	0.0349
375	0.43	0.0300
675	0.77	0.0349
1050	1.19	0.0700
1425	1.62	0.1199
1800	2.04	0.4600
2175	2.47	0.0400
2475	2.81	0.0349
3375	3.83	0.0300

FIGURE 1

© Copyright 2001 Technical Associates of Charlotte, P.C.

IMPROVEMENT OF VIBRATION FOLLOWING ALIGNMENT & BALANCING OF A SPENCER VACUUM PUMP

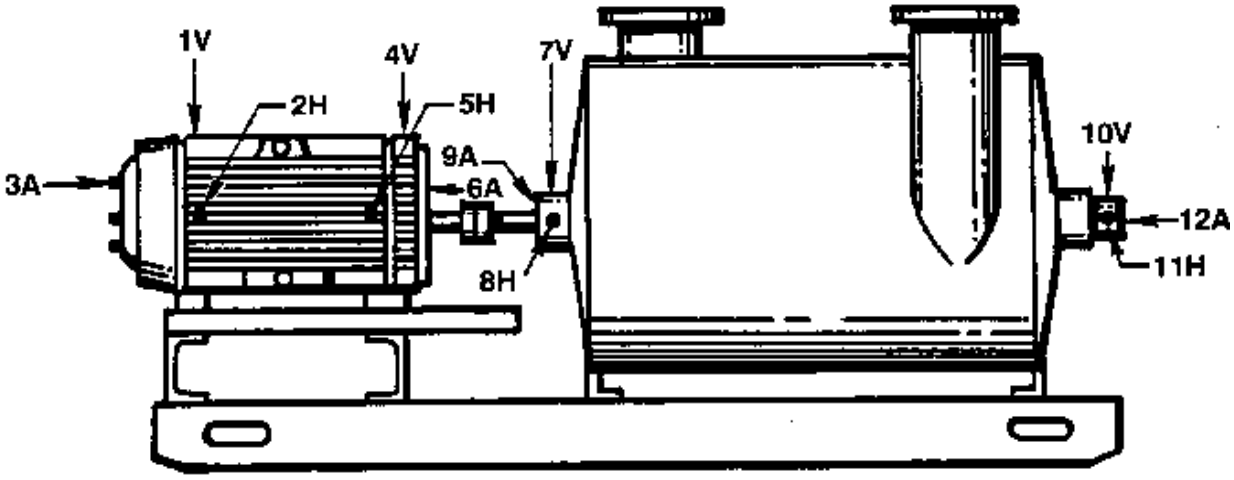
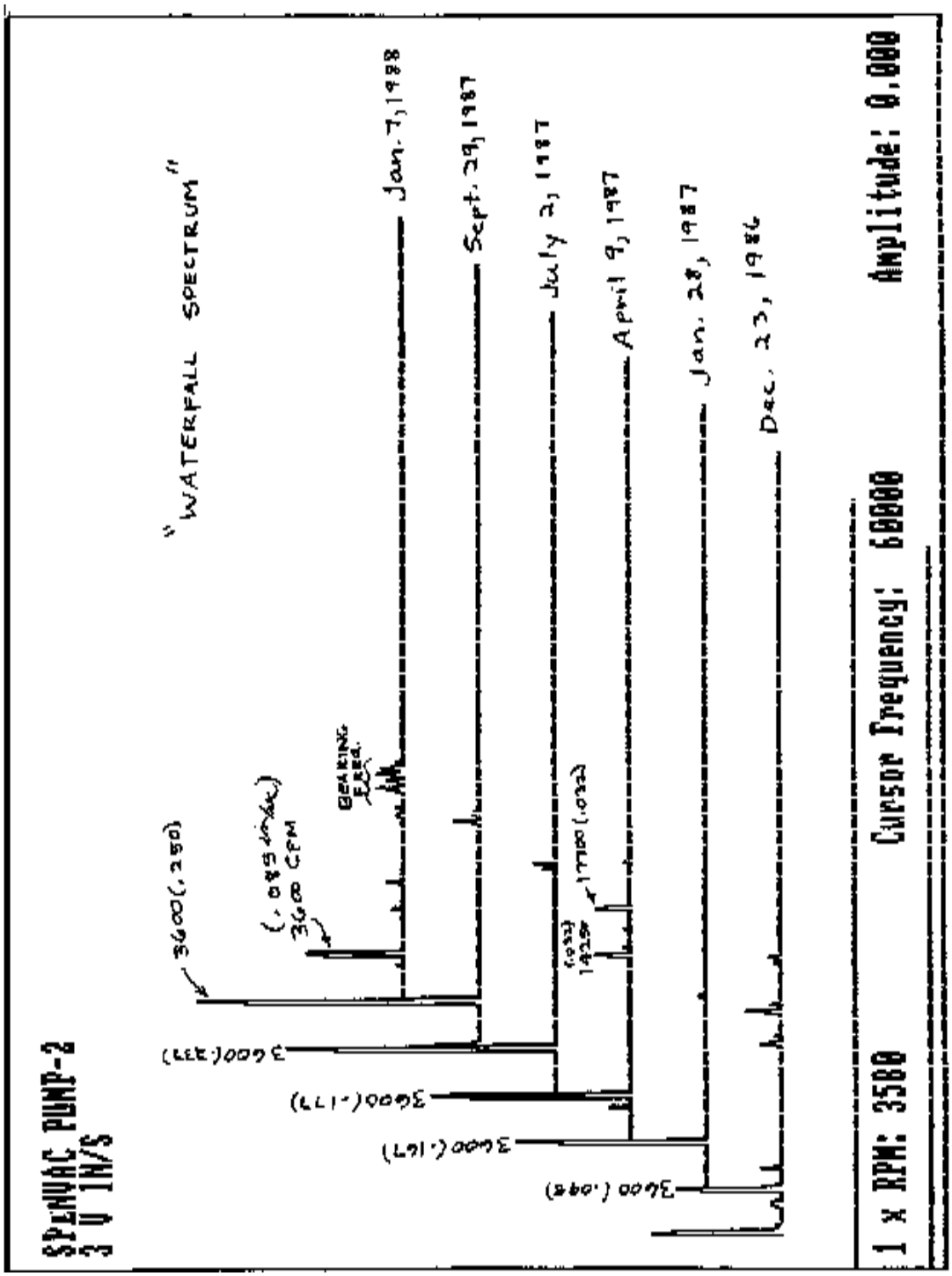


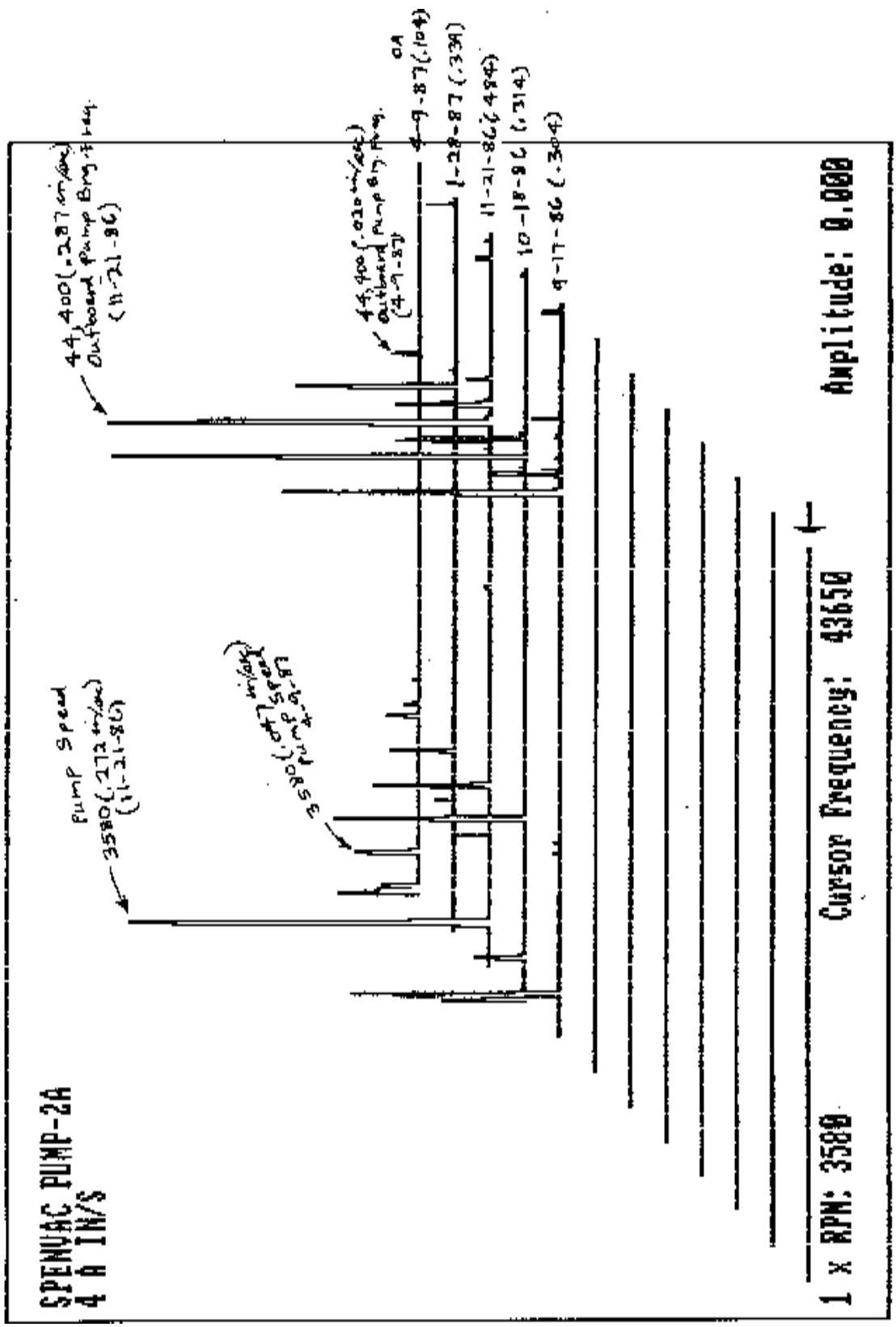
FIGURE 1

© Copyright 2001 Technical Associates of Charlotte, P.C.



IMPROVEMENT OF MACHINE VIBRATION SIGNATURE
FOLLOWING ALIGNMENT OF MOTOR AND VACUUM PUMP SHAFTS
FIGURE 2

© Copyright 2001 Technical Associates of Charlotte, P.C.



SAMPLE 6. REDUCTION IN VIBRATION FOLLOWING BEARING REPLACEMENT AND BALANCING OF VACUUM PUMP (USING IN-PLACE BALANCE WHEELS)

FIGURE 3

© Copyright 2001 Technical Associates of Charlotte, P.C.

CORRECTION OF INCINERATOR FAN WITH A LOOSE BASE AND UNBALANCE (LOOSENESS TYPE A - 1X RPM)

I. BACKGROUND

A client's Predictive Maintenance Program measurements detected high vibration levels at their McGill incinerator fan shown in Figure 1. They diagnosed the problem as fan unbalance and requested that Technical Associates "balance" the fan. The customer did not have balancing capabilities and previous attempts by other outside vendors had not produced satisfactory results. The fan was integrally mounted, overhung fashion, onto the motor shaft as illustrated in Figure 1. The motor rotated at 3574 RPM.

II. RESULTS

Vibration spectral and phase diagnostics were performed before making any balance attempts. The resulting vibration measurements showed the overall vertical vibration at Position 2V had the highest amplitude at .452 in/sec as can be seen in the "BEFORE" column of Table I. Looking at Figure 1 and Table II, vertical phase measurements at Positions 1, 3, 4 and 5 revealed in-phase motion between Positions 1 and 3, in-phase motion between Positions 3 and 4, but a major phase change of 145° between Positions 4 and 5 (that is, between the motor baseplate and the concrete foundation). The horizontal overall vibration at Position 2H was only .228 in/sec. Spectra for motor Positions 2H and 2V were dominated by 1X RPM vibration as seen in the "BEFORE" spectra of Figures 2A and 3A indicating the likelihood of unbalance existed (.028 in/sec at 2H while 2V had .430 in/sec at 1X RPM). However, since a vertical vibration much higher than a horizontal vibration at 1X RPM is often indicative of a loose/weak/broken base, and because of the vertical phase anomaly, a visual inspection was performed looking for this possibility. The inspection found the anchor bolt nut at Position 6 on Figure 1 was rusted (frozen) onto the anchor bolt threads in a "backed off" condition. Thus at Position 6, the base was able to vibrate freely in a vertical direction. After the nut was freed by heating, it was tightened securely against the base flange. Figures 4A and 4B show the 1X RPM vertical vibration at Position 3 on the motor base where 1X RPM vibration fell from .255 in/sec to .108 in/sec "AFTER" tightening (a 58% reduction before any balancing was performed). This was a significant improvement. Without tightening this nut, the ensuing balance would likely have been difficult and unfruitful.

The fan balance then proceeded in a normal fashion from this point on, reducing vibration levels as noted in the "AFTER" column of Table I. Position 2H overall vibration decreased to .099 in/sec from .228 in/sec and the 2V vibration to .144 in/sec from .452 in/sec. The "AFTER" balancing spectra for Positions 2H and 2V are shown in Figures 2B and 3B, respectively. Notice in the Position 2H and 2V spectra of Figures 2B and 3B that little effect was seen at 7200 CPM as a result of the bolt tightening and fan wheel balance. This was due to the 7200 CPM frequency peak being composed of both a 2X RPM component at 7148.7 CPM and a 2X electrical line frequency (2X 3600 CPM) component at 7200 CPM as seen in the "zoom" spectra of Figure 5 taken on Position 2V before the bolt tightening and balance. Therefore, in this case, tightening loose bolts and balancing did not lower vibration from an electrical source at 7200 CPM.

© Copyright 2001 Technical Associates of Charlotte, P.C.

III. CONCLUSIONS

A. The “unbalance” problem was actually a combination of looseness and unbalance.

B. Though the “unbalance” vibration appeared to lessen when an anchor bolt was tightened, it was actually the looseness that was being reduced, and not the unbalance.

C. Tightening bolts and balancing did not correct slight electrical problems.

IV. RECOMMENDATIONS

A. Investigate all “unbalance” problems prior to any balance attempts. Often, additional conditions will be detected that must be corrected prior to balancing. Balancing is usually the last corrective measure taken.

B. It was also recommended that the slight electrical problem likely due to slight stator problems be trended in future spectra taken on this machine.

CF1 R-9208-1

© Copyright 2001 Technical Associates of Charlotte, P.C.

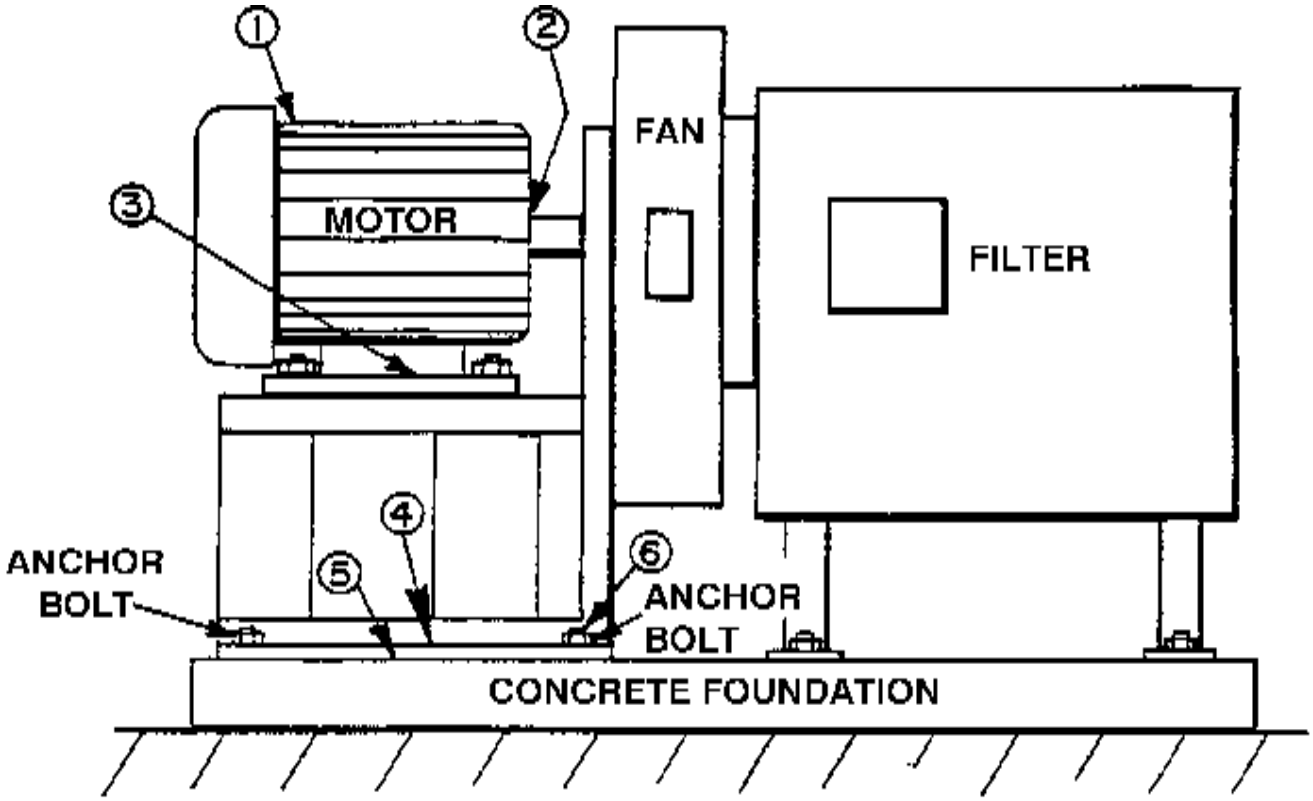


FIGURE 1

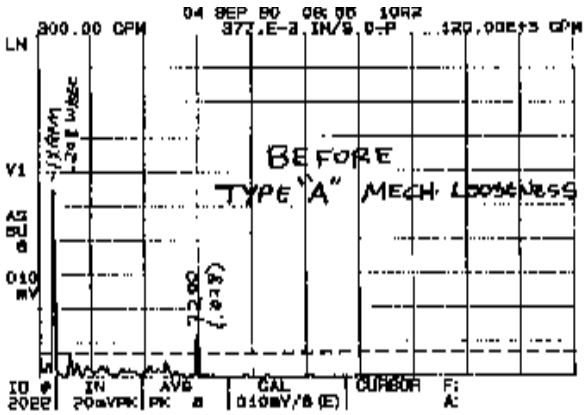
TABLE I. AMPLITUDES BEFORE AND AFTER BALANCING INCINERATOR FAN

POSITION	PHASE MEAS.	OVERALL (IN/SEC)			1 X RPM (IN/SEC)		
		BEFORE	AFTER	% CHANGE	BEFORE	AFTER	% CHANGE
1A	20°	.193	.140	- 27%	.155	.090	- 42%
1H	5°	.410	.196	- 52%	.392	.172	- 56%
1V	85°	.203	.137	- 33%	.145	.072	- 50%
2A	30°	.324	.148	- 54%	.306	.126	- 59%
2H	10°	.228	.099	- 57%	.208	.057	- 73%
2V	95°	.452	.144	- 68%	.430	.103	- 76%

TABLE II. VERTICAL PHASE ANALYSIS BEFORE CORRECTIVE ACTIONS

COMPONENT DESCRIPTION	POS. NO.	VERTICAL PHASE	EVALUATION
Top of Motor	1V	85°	Reference Phase
Motor Support Frame	3V	95°	OK (In-Phase)
Motor Baseplate	4V	110°	OK (In-Phase)
Concrete Foundation	5V	255°	145° Phase Change (Excessive)

© Copyright 2001 Technical Associates of Charlotte, P.C.



IN/S 0-P	CPM
208.E-3 -1X	3600.0
27.5E-3	7200.0
46.5E-3	35700.

PEAK PICK ON TRACE 1

O/A .228 in/sec

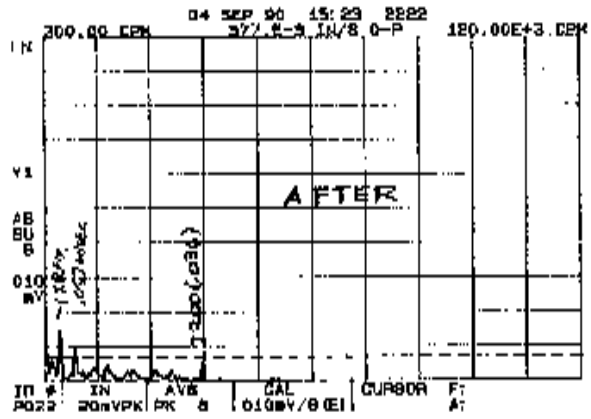
POS. 2 HOR-BEFORE TIGHTENING BOLTS AND BALANCING

05-SEP-90

09: 38: 08

FIGURE 3A

MCGILL FAN



IN/S 0-P	CPM
56.8E-3 -1X	3600.0
36.1E-3	7200.0

PEAK PICK ON TRACE 1

O/A .099 in/sec

POS. 2 HOR-AFTER TIGHTENING BOLTS AND BALANCING

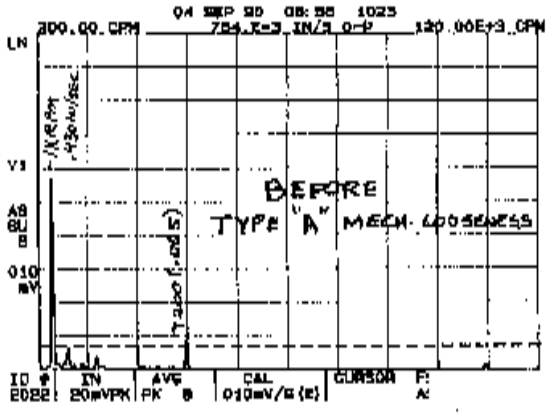
05-SEP-90

09: 58: 58

FIGURE 3B

MCGILL FAN

© Copyright 2001 Technical Associates of Charlotte, P.C.



IN/S 0-P	CPM
430.E-3-1X	3600.0
65.4E-3	35700.

O/A .452 In/sec

PEAK PICK ON TRACE 1

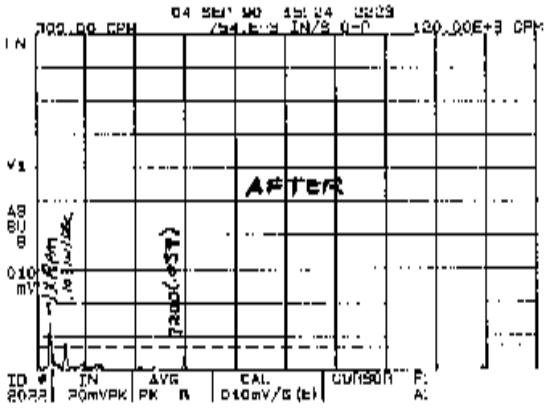
POS 2 VERT-BEFORE TIGHTENING BOLTS AND BALANCING

05-SEP-90

09: 39: 25

FIGURE 4A

MCGILL FAN



IN/S 0-P	CPM
103.E 3-1X	3600.0
58.5E-3	7200.0

O/A .144 In/sec

PEAK PICK ON TRACE 1

POS .2 VERT-AFTER TIGHTENING BOLTS AND BALANCING

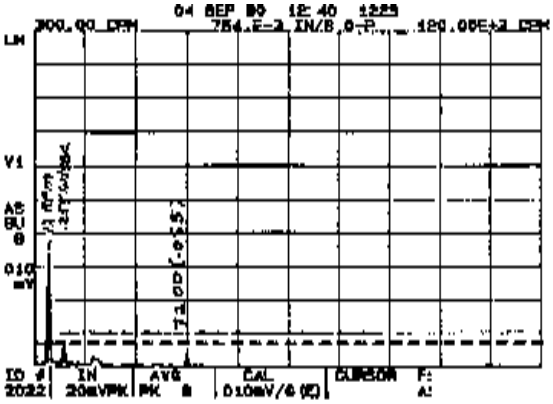
05 SEP-90

10: 42: 51

FIGURE 4B

MCGILL FAN

© Copyright 2001 Technical Associates of Charlotte, P.C.



IN/S 0-P	CPM
255.E-3-1X	3600.0
55.1E-3	7200.0

PEAK PICK ON TRACE 1

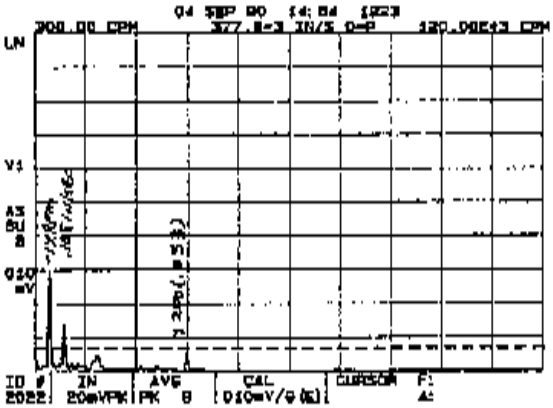
O/A .275 In/sec
POS 3 VERT-BEFORE BOLTS TIGHTENED AND BEFORE BALANCING

05-SEP-90

FIGURE 5A

09: 50: 25

Position 3-V
MCGILL FAN



IN/S 0-P	CPM
108.E-3-1X	3600.0
53.4E-3	7200.0

PEAK PICK ON TRACE 1

O/A .135 in/sec
POS 3 VERT-AFTER BOLTS TIGHTENED AND BEFORE BALANCING

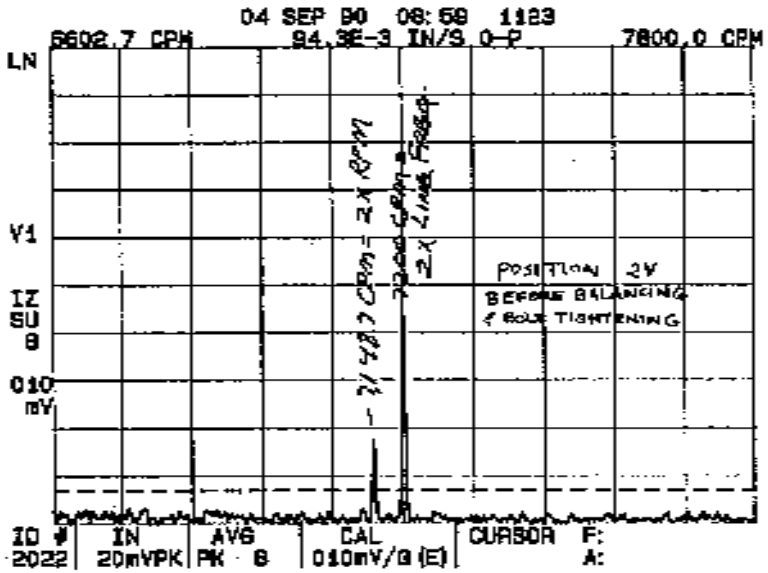
05-SEP-90

FIGURE 5B

09: 51: 36

Position 3-V
MCGILL FAN

© Copyright 2001 Technical Associates of Charlotte, P.C.



PEAK PICK ON TRACE 1

IN/S 0-P		CPM
16.4E-3	2X RPM	7148.7
40.9E-3	2X ELECTRICAL LINE FREQ. (120 HERTZ)	7200.0

05-SEP-90

09:41:51

FIGURE 6

'ZOOM' SPECTRUM AT
POSITION 2V SEPARATING
2X RPM (7,148.7 CPM) FROM 2X ELECTRICAL
LINE FREQUENCY (2X 3600 CPM = 7200 CPM)

POSITION 2-V

MCGILL FAN

© Copyright 2001 Technical Associates of Charlotte, P.C.

UNBALANCE AND LOOSENESS PROBLEMS ON AN IN-LINE INDUCED DRAFT FAN

(LOOSENESS TYPE B FROM DIAGNOSTIC CHART)

I. BACKGROUND

A manufacturer of in-line induced draft fans used in HVAC ducts wanted to see if vibration levels in their production models could be reduced through balancing. The fan had a small paddle wheel impeller mounted directly on the shaft of a fractional horsepower motor in the configuration illustrated in Figure 1. The unit is mounted to ductwork when in use.

II. RESULTS

The initial fan vibration spectra showed an unbalance at 1X RPM of .250 in/sec along with .052 in/sec of looseness at 2X RPM as seen in Figure 2. The overall vibration was .256 in/sec at this time. After the addition of 1.2 grams of balance correction weight, the unbalance was successfully reduced to .053 in/sec - a 79% reduction. Figure 3 pictures the “After” spectra. The “After” looseness at 2X RPM was .089 in/sec, approximately the same as before the balance. The overall vibration was now .108 in/sec - a 58% reduction. The balance was a success. The looseness problem manifested at 2X RPM was relatively unaffected.

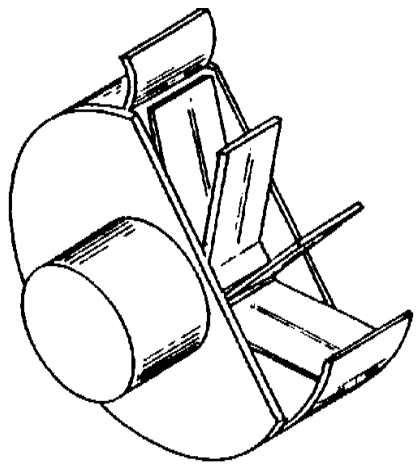


FIGURE 1
IN-LINE INDUCED DRAFT FAN

© Copyright 2001 Technical Associates of Charlotte, P.C.

III. CONCLUSIONS

A. Correcting unbalance does not remove a pre-existing looseness condition, or practically any other position for that matter. Balancing may improve other existing conditions since it reduces the amount of vibration energy but, it does not solve them. Even at the supposed "unbalance" frequency of 1X RPM, other conditions may exist that will limit the improvements achievable through balancing. Balancing is done only after all other conditions (i.e., misalignment, looseness, etc.) are addressed.

B. The overall vibration in this machine did not improve as much as did the 1X RPM vibration. This results from the premise that correcting unbalance does not correct other conditions that also affect the overall vibration levels. Further reduction in the overall vibration level would require correcting the looseness condition.

C. The final balance weight was 1.2 grams, about half the weight of a dime or two No. 1 paper clips. This is an indication of the system's sensitivity. In this case, it may not take long for dirt to build up a 1.2 gram unbalance. Extreme sensitivity often results from looseness, resonance or a weak mounting structure. It is desirable to reduce the sensitivity levels in extremely sensitive machines.

IV. RECOMMENDATIONS

A. It was recommended that the source of the 2X RPM looseness vibration be identified. The client was told to look closely for possible cracks in the frame, loose bolts or possibly even misaligned (cocked) bearings. Correction of these problems would likely greatly reduce vibration at 2X RPM.

B. A balancing stand was recommended for the plant to enable them to balance the fan wheels right off the production line. A balance tolerance of ISO G-2.5 was recommended to provide satisfactory balance with vibration at 1X RPM likely below approximately .075 in/sec peak velocity.

CF2 R-1092-1

© Copyright 2001 Technical Associates of Charlotte, P.C.

FIGURE 2

FIGURE 3

© Copyright 2001 Technical Associates of Charlotte, P.C.

DETECTION OF FAN WHEEL ROTOR LOOSENESS ON A NUVA FEEDER VENT FAN

I. BACKGROUND

A utility company contracted Technical Associates to establish and install a predictive maintenance vibration program at each of their fossil fuel plants. Additionally, the client wanted Technical Associates to acquire the baseline measurements and issue a **Baseline Report**. This report would include the diagnoses of all problems found ranking these problems in order of severity, a condition evaluation of each machine tested, a drawing of each machine evaluated showing the locations where measurements were acquired and a plant layout showing the location of each machine which had been baselined.

II. DISCUSSION OF RESULTS

There were many machines baselined and many problems were found. One particular problem was found on the Nuva Feeder Vent Fan 68N as seen in the machine sketch Figure 1. The fan had an overall vibration of .460 in/sec (which caused it to exceed its alarm level of .375 in/sec) at Position 4 horizontal as seen in the Machine Report of Figure 2. The Position 4H spectrum, Figure 3, showed some well defined multiple harmonics of running speed with a large 1X RPM amplitude of .340 in/sec. Technical Associates defines this spectral appearance as a Type "C" looseness problem, often attributed to a poor bearing fit on the shaft or in its housing (pillowblock in this case). However, it should be pointed out this same signature can result from a bearing turning on a shaft (a Stage 4 bearing failure condition).

This Type "C" looseness is often the result of a replacement anti-friction bearing being installed on a worn (or undersized) shaft, or in a "wallowed-out" (oversized) bearing housing. A Stage 4 bearing catastrophic failure can occur due to total degradation of the bearing components (i.e., rolling elements and races) finally resulting in an unstable mechanical looseness. Neither the Type "C" nor Stage 4 looseness provide sufficient constraints to keep a shaft on its center of rotation which results in amplification of even a small imbalance.

III. CONCLUSIONS

A. The Position 4 bearing had a mechanical looseness condition that could be a result of bearing fit, bearing wear, bearing turning on a shaft, broken structures and/or loosened fasteners.

B. The unbalance may not have been a true unbalance, but may have resulted from the looseness condition aggravating and amplifying an otherwise insignificant unbalance.

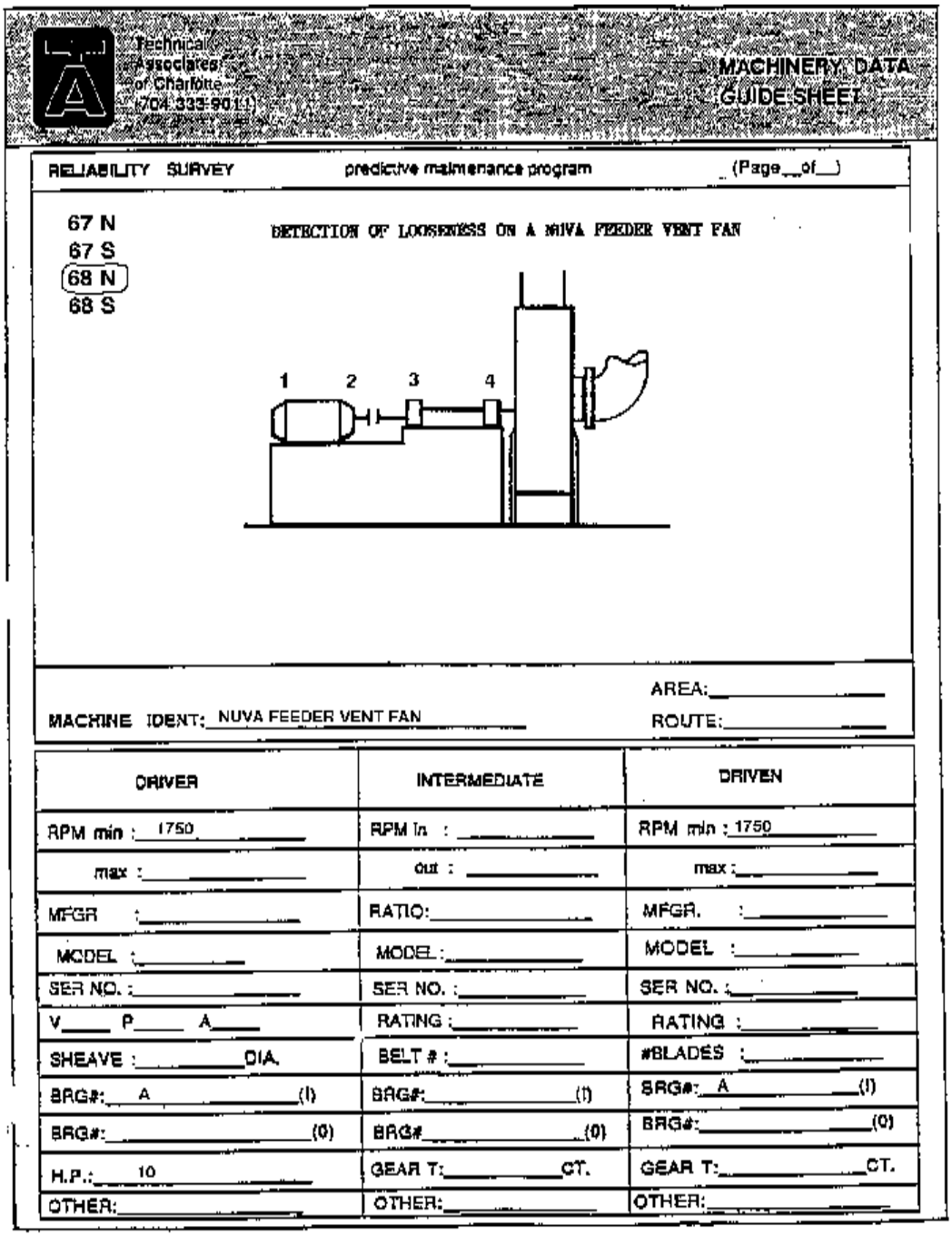
© Copyright 2001 Technical Associates of Charlotte, P.C.

IV. RECOMMENDATIONS

A. Correct the looseness condition.

B. Make a careful field check to ensure neither fan bearing was turning on the shaft.

C. After correcting the looseness condition, re-evaluate the imbalance and balance if necessary.



Technical Associates of Charlotte
(704) 333-9011

MACHINERY DATA GUIDE SHEET

RELIABILITY SURVEY — predictive maintenance program — (Page__of__)

67 N
67 S
68 N
68 S

DETECTION OF LOOSENESS ON A NUVA FEEDER VENT FAN

MACHINE IDENT: NUVA FEEDER VENT FAN

AREA:

ROUTE:

DRIVER	INTERMEDIATE	DRIVEN
RPM min : 1750	RPM In :	RPM min : 1750
max :	out :	max :
MFGR :	RATIO:	MFGR. :
MODEL :	MODEL :	MODEL :
SER NO. :	SER NO. :	SER NO. :
V___ P___ A___	RATING :	RATING :
SHEAVE : ___DIA.	BELT # :	#BLADES :
BRG#: A (I)	BRG#: (I)	BRG#: A (I)
BRG#: (O)	BRG# (O)	BRG#: (O)
H.P.: 10	GEAR T: ___CT.	GEAR T: ___CT.
OTHER:	OTHER:	OTHER:

DCB-17H

FIGURE 1

CF3 R-1092-1

© Copyright 2001 Technical Associates of Charlotte, P.C.

IRD Mechanalysis Data Analysis System
Dec-04-89

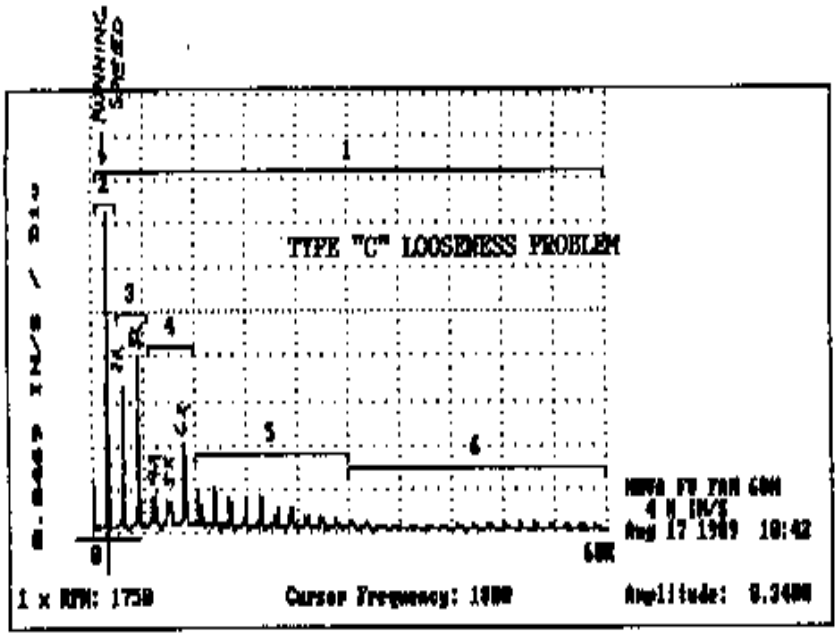
Machine Report (Latest Data) for: NUVA FV FAN 6BN

Date	Time	Pos	Dir	Amol	Units	Change	Alarm	Inspection
Aug-17-89	10:39	1	H	.190	IN/S	--------%	.375	
Aug-17-89	10:39	1	H	.042	g/SE	--------%	.750	
Aug-17-89	10:39	1	A	.132	IN/S	--------%	.375	
Aug-17-89	10:40	1	V	.092	IN/S	--------%	.375	
Aug-17-89	10:40	2	H	.130	IN/S	--------%	.375	
Aug-17-89	10:40	2	H	.256	g/SE	--------%	.750	
Aug-17-89	10:40	2	V	.089	IN/S	--------%	.375	
Aug-17-89	10:41	3	A	.309	IN/S	--------%	.375	
Aug-17-89	10:41	3	H	.132	IN/S	--------%	.375	
Aug-17-89	10:41	3	H	0.72	g/SE	--------%	.750	
Aug-17-89	10:41	3	V	.177	IN/S	--------%	.375	
Aug-17-89	10:42	4	A	.296	IN/S	--------%	.375	
Aug-17-89	10:42	4	H	*.460	IN/S	--------%	.375	
Aug-17-89	10:42	4	H	.229	g/SE	--------%	.750	
Aug-17-89	10:43	4	V	.240	IN/S	--------%	.375	

15 lines this report

FIGURE 2

© Copyright 2001 Technical Associates of Charlotte, P.C.



002% x 0.500 = 0.0100 1 x RPM: 1750.00

Machine	Pos	Dir	Units	Date
NUVA FV FAN 68N	4	H	IN/S	Aug 17 1989 10:42

Frequency	1 x RPM	Amplitude	Frequency	1 x RPM	Amplitude
150	0.09	0.0525	19800	11.31	0.0375
1800	1.03	0.3400	21300	12.17	0.0175
3600	2.06	0.1575	21600	12.34	0.0275
5400	3.09	0.1075	23100	13.20	0.0275
6900	3.94	0.0375	23400	13.37	0.0275
7350	4.20	0.0450	24750	14.14	0.0175
8700	4.97	0.0350	25200	14.40	0.0175
9150	5.23	0.0325	26550	15.17	0.0175
10650	6.09	0.0975	27000	15.43	0.0150
12300	7.03	0.0475	28350	16.20	0.0150
12600	7.20	0.0300	30150	17.23	0.0125
14100	8.06	0.0500	30450	17.40	0.0125
14400	8.23	0.0425	32100	18.34	0.0125
15900	9.09	0.0375	48000	27.43	0.0125
16200	9.26	0.0325	49800	28.46	0.0125
17850	10.20	0.0375	51600	29.49	0.0125
19500	11.14	0.0400			

T - Threshold adjust Q - Quit

FIGURE 3

© Copyright 2001 Technical Associates of Charlotte, P.C.

DETECTION OF SERIOUS WEAR OF A CENTRIFUGAL COMPRESSOR BULL GEAR SKF 6213 BALL BEARING

I. BACKGROUND

An annual contract was awarded Technical Associates beginning January, 1984 encompassing 52 machines including several Ingersoll-Rand Centac centrifugal air compressors. The primary instrument used at that time was a Hewlett Packard model 3561A real-time analyzer. Measurements were taken at 4-week intervals on each of the machines. Figures 1 thru 3 include drawings showing one of the Ingersoll-Rand Centac compressors (in this case, a model C-21 having an output capacity of 2100 ICFM). Figure 1 shows that measurements were taken on each of the motor bearing housings as well as on the bullgear (Position 3) and each of the four impeller rotor bearing housings (Positions 4 thru 7). Note the pictorial view of the bullgear and each of its mating pinions on Figure 1. The bullgear operated at about 3580 RPM (59.67 Hz), whereas the impeller rotor speeds ranged from 29,325 RPM (488.75 Hz) for the 1st Stage up to 49,270 RPM (821.13 Hz) in the case of the 4th Stage.

The problem with these machines is that once a defect occurs on either a bullgear or on one of its mating pinions, it can propagate rapidly from one component to another. Note that the cost for rebuilt rotors at that time (1984) ranged from about $8,000 for the 4th Stage up to around $26,000 for the 1st Stage. Thus, it was most important to detect problems in initial stages and take corrective actions before the problems were allowed to transmit from one component to another.

Figure 2 shows a typical cross section of one of the impeller rotors, each of which were outfitted with proximity probes which were serving permanent vibration monitors. The Alarm level for these machines was set at .70 mil (Pk-Pk) whereas the Trip level was at 1.0 mil (as per the manufacturer).

It is important to point out that each of three types of measurements were taken on these machines including:

A. **Vibration Displacement Measurements Directly from Proximity Probe Monitors** - a cable is connected directly to the outputs of each of the four proximity probes serving each of the four impeller stages. This cable is then connected directly to the HP 3561A analyzer allowing the capture not only of overall displacement (which is displayed by the monitor), but also enables the generation of complete vibration spectra of the shaft vibration at each of these stages.

B. **Vibration Velocity Measurements from each Bearing Housing** - a standard accelerometer having a frequency range of 1.0 to 10,000 Hz (60 - 600,000 CPM) is mounted on each of the bearing housings in the locations shown in Figure 1. A number of different frequency range spectra are captured on the various bearing housings, depending on the speed of the shaft at that location. The acceleration signal from the accelerometer is integrated to velocity within the HP 3561A analyzer (from peak acceleration to peak velocity).

© Copyright 2001 Technical Associates of Charlotte, P.C.

C. **High Frequency Acceleration Measurement from Compressor Bearing Housings** - a special, high frequency accelerometer is employed along with signal conditioning instrumentation in order to capture measurements up to at least 2 X gear mesh frequency (2 GMF). Table I shows that the fundamental gear mesh frequency (GMF) is 20,528 Hz (1,231,700 CPM). Because measurements need to be taken up to 2 and 3 X gear mesh frequency (2 GMF and 3 GMF), this puts great demands on the transducer system as well as the mounting of the accelerometer onto the housing as well. Great attention to each of these items has enabled measurements with repeatable amplitudes up thru 3 GMF (61,585 Hz) although the absolute value of the amplitude itself at this high frequency might be in question. That is, if the accelerometer is mounted and measurements taken several different times, it is possible to get good repeatable acceleration measurements. However, if the system displays 10 g acceleration, the true level might be 7 or 8 g, or on the other hand, it might be 12 to 14 g. In any case, due to the good repeatability of the measurement, data taken from one month to the next is successfully trended. In fact, these same procedures were used on approximately 30 different Centac compressors on a monthly basis during 1984 by Technical Associates and have been used on several hundred additional Centacs since that time.

II. ANALYSIS RESULTS

Figure 3 includes a section view showing where measurements are taken in the axial direction on the bullgear (Position 3A), 2nd Stage (Position 5A) and 3rd Stage (Position 6A). Note the location of each of the bullgear ball bearings in Figure 3. The SKF 6213 bearing is inboard (closest to the coupling) whereas the SKF 7213 is somewhat remote from the measurement location. Note that the vibration signal from the SKF 6213 bearing transmits cleanly from the housing through the casing into the casing bolt on which the accelerometer is mounted. On the other hand, the vibration from the SKF 7213 bearing must travel along the shaft and then through either the SKF 6213 bearing and/or machine casing. Therefore, a number of experiments over the years on these machines have shown suppressed amplitudes which must be taken into account that come from bearing defect frequencies from the SKF 7213 bearing relative to those of the SKF 6213.

Table I provides many of the anticipated vibration frequencies generated by components of the compressor. These include the gear mesh frequency (GMF), the blade pass frequencies of each of the 4 Stages (BPF), each of the impeller rotor speeds (F_1 thru F_4), the main oil pump frequency (F_{PUMP}), the SKF 7213 bearing defect frequencies, the SKF 6213 bearing defect frequencies, and finally the bullgear and motor running speed frequency (F_M). Note that this table includes the first 10 harmonics of each of these frequencies (1F thru 10F).

Figure 4 shows one of these spectra captured approximately 9 months after initial baselines were captured. Note that several frequencies appear on this spectrum including the 2nd Stage running speed peak (F_2), the 3rd Stage speed (F_3) and the 4th Stage speed (F_4). Also showing up was a high amplitude peak at approximately 352 Hz. Referring back to Table I, note that the 352 Hz frequency corresponds to the inner race defect frequency of the SKF 6213 bearing (6213 BPFI).

© Copyright 2001 Technical Associates of Charlotte, P.C.

Importantly, when initial baseline signatures were captured in January, 1984, this 352 Hz peak appeared with an amplitude of .216 in/sec and did not have bullgear running speed sidebands to its left and right (which later did appear as shown in Figure 4). Due to the baseline amplitude of over .20 in/sec which is quite high for a bearing defect frequency, Technical Associates recommended replacement of the bullgear bearing immediately before the problem, then isolated only with the bullgear bearing, caused problems with the mating components. Due to great demands on this compressor and its support of critical process machinery, the plant decided it could not shut down the machine to accomplish the bearing replacement at this time. Therefore, the decision was made to simply trend future spectra on this machine and try to schedule this replacement along with the production department ASAP. Three months later, the amplitude of this frequency began climbing and went up to about .252 in/sec as of April, 1984. At this point, small amplitude sideband frequencies at bullgear speed began to appear to the left and right of this bearing frequency. Once again, Technical Associates recommended bullgear bearing replacement even though the other measurements on this machine did not indicate any damage to other machine components yet (for example, acceleration measurements still showed that no gear mesh frequencies exceeded about 7 g which is normal for these machines at these GMF frequencies and harmonics). Still the plant felt it could not schedule the bearing replacement.

By June, amplitudes at BPFI (352 Hz) increased from .252 in/sec in April up to .305 in/sec and sidebands at 292 and 410 Hz were now well developed with amplitudes of almost .05 in/sec each. Because amplitudes at gear mesh frequencies had now begun to increase from about 7 g in both January and April up to approximately 12 g now in June, Technical Associates recommended immediate bullgear bearing replacement before significant damage was done to the impeller rotors. However, once again, the plant production department would not allow the machine to be brought down for maintenance.

Therefore, signatures later captured on September 28 showed extensive problems throughout the compressor. Figure 4 shows that the BPFI amplitude was now up to .402 in/sec and had well developed running speed sidebands at 292.5 Hz and 410.0 Hz. In addition, it now showed a small second harmonic inner race frequency at 705 Hz. Of even greater concern were the acceleration spectra captured on the bullgear and pinion housings which are shown in Figures 6 and 7. Figure 6 was a wideband plot from 0 to 100,000 Hz (6,000,000 CPM) and showed that levels at twice gear mesh frequency (2 GMF) had now climbed from only 7 g at baseline up to 42.1 g. Then, looking at the zoom spectrum shown in Figure 7, note that 2 GMF was now surrounded by well developed sidebands at 1st Stage speed indicating significant wear of the 1st Stage pinion. Subsequent measurements on the other stages indicated problems with at least three of the pinions as well as the bullgear itself.

III. RECOMMENDATIONS MADE

As of the September 28 data, Technical Associates recommended replacement of each of the bullgear bearings. Also, it was recommended that each of the gears be closely inspected for suspected wear.

© Copyright 2001 Technical Associates of Charlotte, P.C.

IV. CORRECTIVE ACTIONS TAKEN

When inspection of the gears was made, significant wear was found on the bullgear as well as the 1st, 3rd and 4th Stage pinions. As a result, each of these impeller rotor pinion assemblies had to be replaced as well as the bullgear and its bearings. Figures 5 and 8 were captured after corrective actions were taken. Figure 5 shows that the velocity spectrum showed a dramatic drop in inner race frequency from .402 in/sec down to .015 in/sec after replacement of the SKF 6213 bearing. Figure 8 shows that acceleration levels at 2 GMF dropped from 42.1 g before to 7.62 g after replacement of 4 of the 5 gears. One of the hard lessons learned from this case history was that corrective actions should be taken on these centrifugal compressors when vibration spectra indicate if this is at all possible. What would have been less than a $10,000 repair cost had actions been taken right after the January baseline ended up being repair costs of slightly over $100,000 since the problem was allowed to propagate to so many other components. Fortunately, after this case history, the plant has paid close attention to vibration spectra and successfully detected each of 3 potentially serious incidents that could have resulted in similar costly failures. However, on each of these occasions, the maintenance department coordinated with the production department and took actions which prevented any of the failure scenario costs from exceeding $20,000.

© Copyright 2001 Technical Associates of Charlotte, P.C.

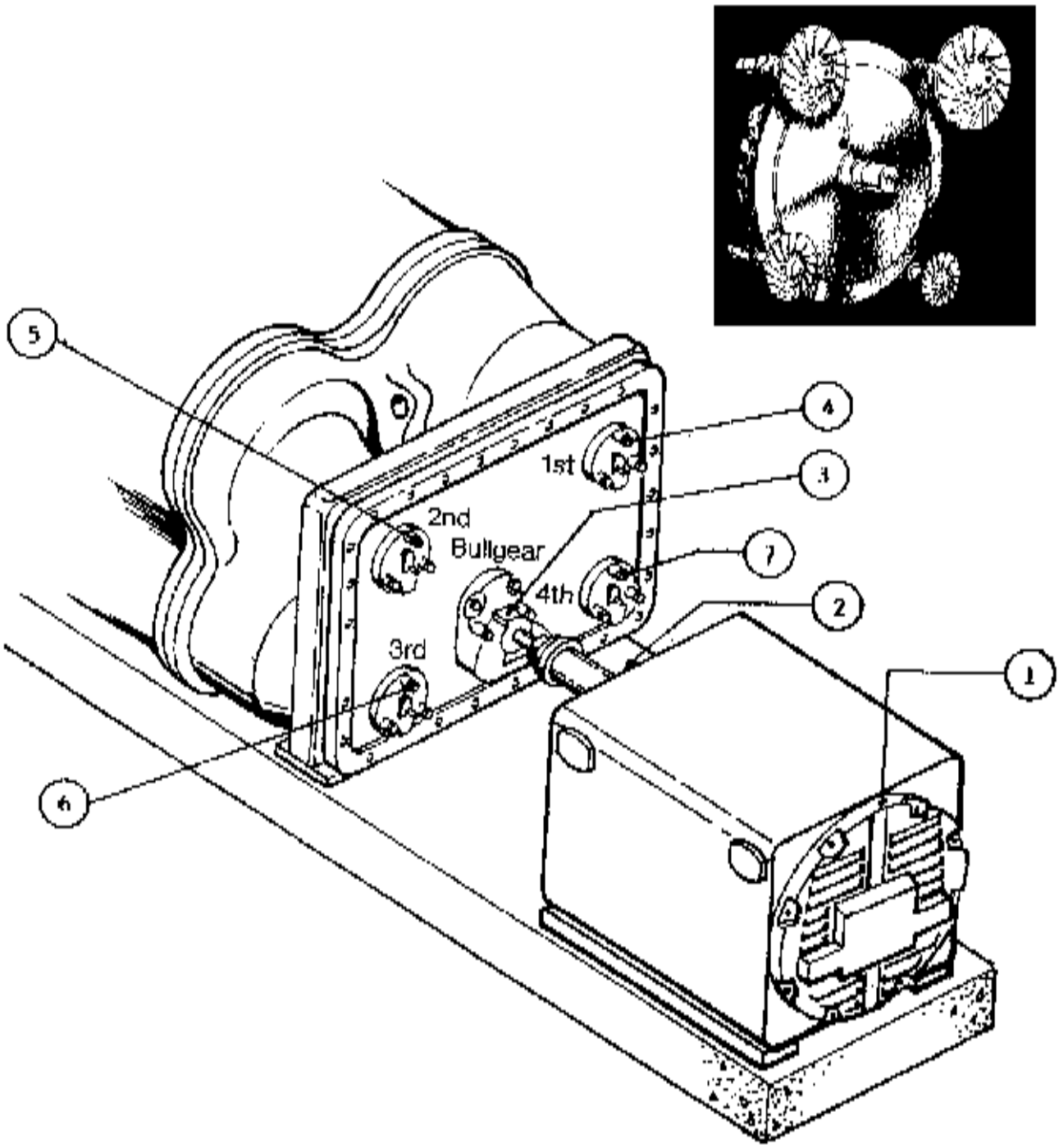
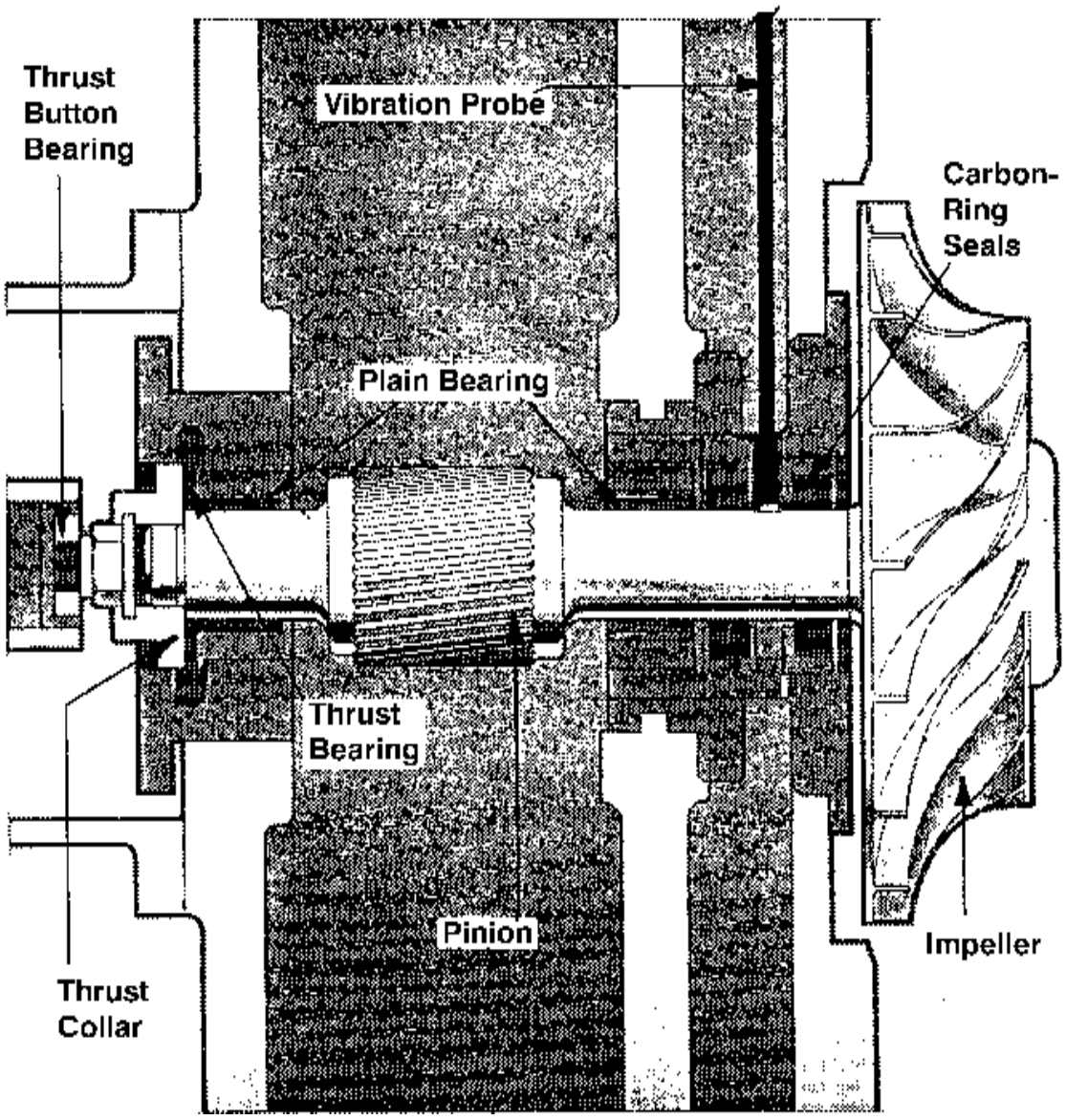


FIGURE 1: ACCELEROMETER LOCATIONS ON AN INGERSOLL-RAND CENTAC CENTRIFUGAL AIR COMPRESSOR (MODEL C-21 2100 ICFM)

© Copyright 2001 Technical Associates of Charlotte, P.C.

FIGURE 2. INGERSOLL-RAND CENTAC IMPELLER ROTOR CROSS SECTION



© Copyright 2001 Technical Associates of Charlotte, P.C.

FIGURE 3. SECTION VIEW OF AN INGERSOLL-RAND CENTAC CENTRIFUGAL AIR COMPRESSOR (FRAME 1)

© Copyright 2001 Technical Associates of Charlotte, P.C.

© Copyright 2001 Technical Associates of Charlotte, P.C.

TABLE 1. ANTICIPATED FREQUENCIES FOR MACHINE #6 C-21 CENTAC (HERTZ)

ITEM FREQ.	$\frac{F_X}{F_M}$	1F	2F	3F	4F	5F	6F	7F	8F	9F	10F
GMF.....	$344F_M$	20,528	41,056	61,585	82,113	102.641	N/A	N/A	N/A	N/A	N/A
BPF_4.....	178.88F	10,675	21,349	32,024	42,699	53,373	64,048	74,723	85,397	96,072	106,747
BPF_3.....	165.63F	9,884	19,768	29,652	39,536	49,420	59,304	69,188	79,072	88,956	98,839
BPF_2.....	150.50F	6,981	17,962	26,943	35,924	44,965	53,886	62,868	71,849	80,830	89,811
BPF_1.....	122.86F	7,331	14,663	21,995	29,326	36,658	43,989	51,320	58,652	65,983	73,315
F_4.....	13.760F	821.13	1642.3	2463.4	3284.5	4105.6	4926.8	5747.9	6569.0	7390.2	8211.3
F_3.....	12.741F	760.30	1520.6	2280.9	3041.2	3801.5	4561.8	5322.1	6082.4	6842.7	7603.0
F_2.....	10.750F	641.51	1283.0	1924.5	2566.0	3207.5	3849.0	4490.5	5132.1	5773.6	6415.1
F_1.....	8.190F	488.8	977.5	1466.3	1955.1	2443.8	2932.6	3421.4	3910.1	4398.9	4887.7
F_{pump}	$13F_M$	775.78	1551.6	2327.3	3103.1	3878.9	4654.7	5430.4	6206.2	6982.0	7757.8
7213 BPFI	9.18F	547.81	1095.6	1643.4	2191.2	2739.0	3286.8	3834.6	4382.4	4930.2	5478.0
7213 BPFO	6.84F	408.18	816.36	1224.54	1632.72	2040.9	2449.08	2857.26	3265.44	3673.62	4081.8
7213 BSF	2.71F	161.72	322.4	483.6	644.8	806.0	967.2	1128.4	1289.6	1450.8	1612.0
7213 FTF	0.426F	25.42	50.84	76.26	101.68	127.1	152.52	177.94	203.36	228.78	254.2
6213 BPFI	5.898F	351.96	703.92	1055.88	1407.84	1759.8	2111.76	2463.72	2815.68	3167.64	3519.6
6213 BPFO	4.098F	244.55	489.1	733.65	978.2	1222.75	1467.3	1711.85	1956.4	2200.95	2445.5
6213 BSF	2.680F	159.93	319.86	479.79	639.72	799.65	959.58	1119.51	1279.44	1439.37	1599.3
6213 FTF	0.410F	24.47	48.94	73.41	97.88	122.35	146.82	171.29	195.76	220.23	244.7
* F_M.....	$1F_M$	59.675	119.35	179.03	238.70	298.38	358.05	417.73	477.40	537.08	596.75

* These calculations are based on a motor running speed of 3580 rpm = 59.675 Hz.

NORMAL OPERATING SPEEDS

COMPONENT	SYMBOL	RPM	Hz
4th Stage	F_4	49,270	821.13
3rd Stage	F_3	45,620	760.30
2nd Stage	F_2	38,490	641.51
1st Stage	F_1	29,325	488.75
Motor	F_M	3,580	59.68

* NOTE: See "Terminology" on page 22 for definitions of "ITEM FREQUENCIES" in above Table VIII.

FIGURE 4: SEVERE DEFECT CONCENTRATED ON ONE LOCATION OF THE INNER RACE OF THE SKF 6213 BULLGEAR BEARING (9/28/84)

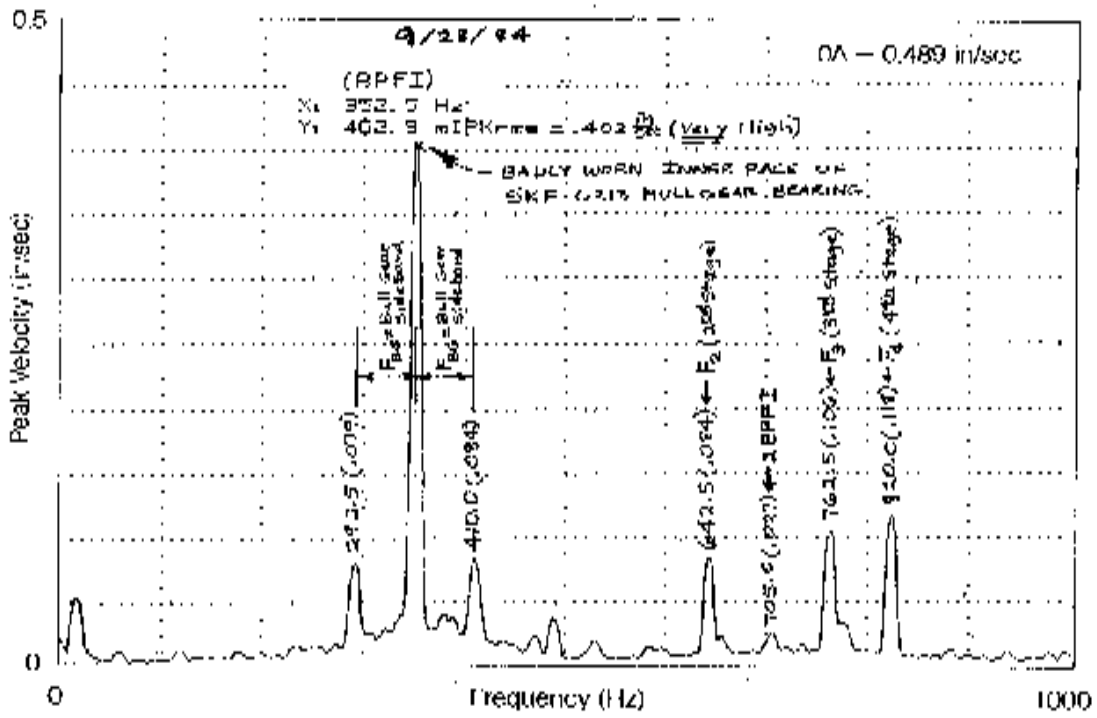
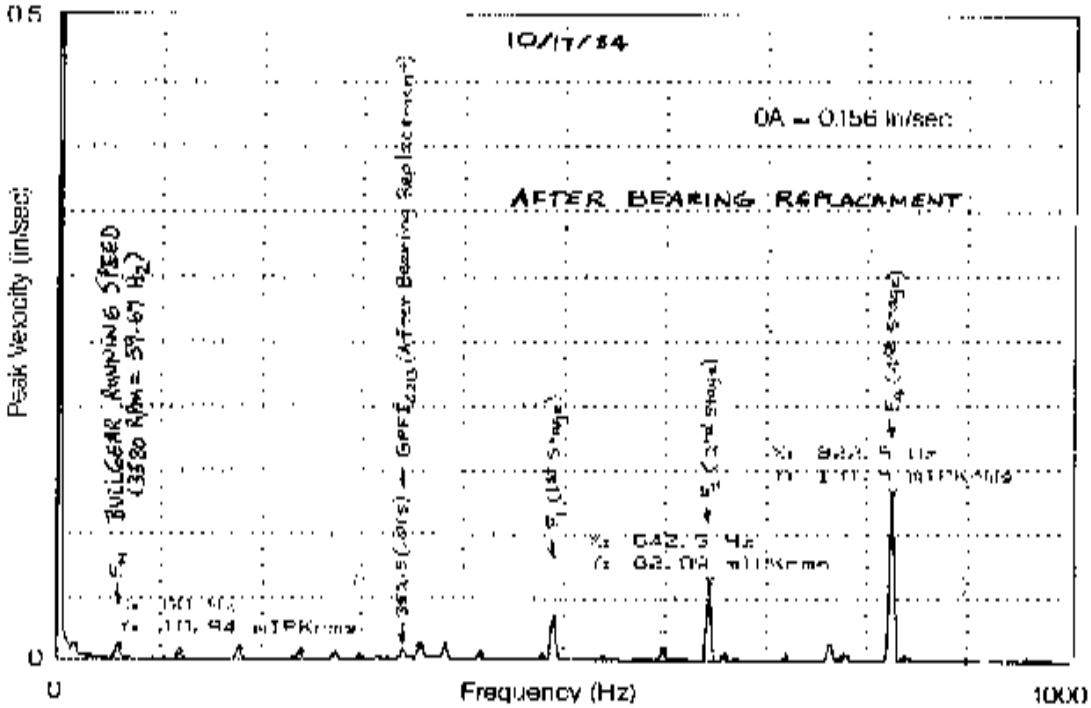


FIGURE 5: AFTER REPLACEMENT OF SKF 6213 BULLGEAR BEARINGS (10/16/84)



INGERSOLL-RAND CENTAC BULLGEAR BALL BEARING CAUGHT JUST PRIOR TO FAILURE

© Copyright 2001 Technical Associates of Charlotte, P.C.

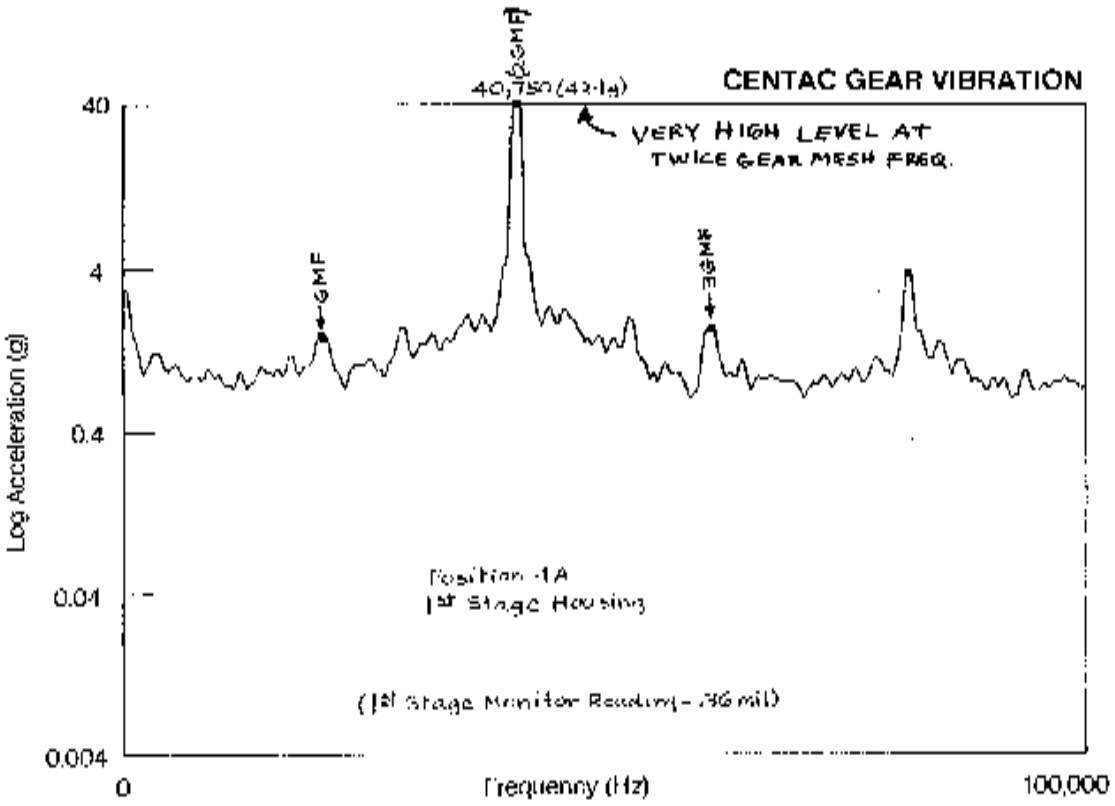


FIGURE 6: HIGH ACCELERATION DUE TO GEAR WEAR 9 MONTHS AFTER BASELINING

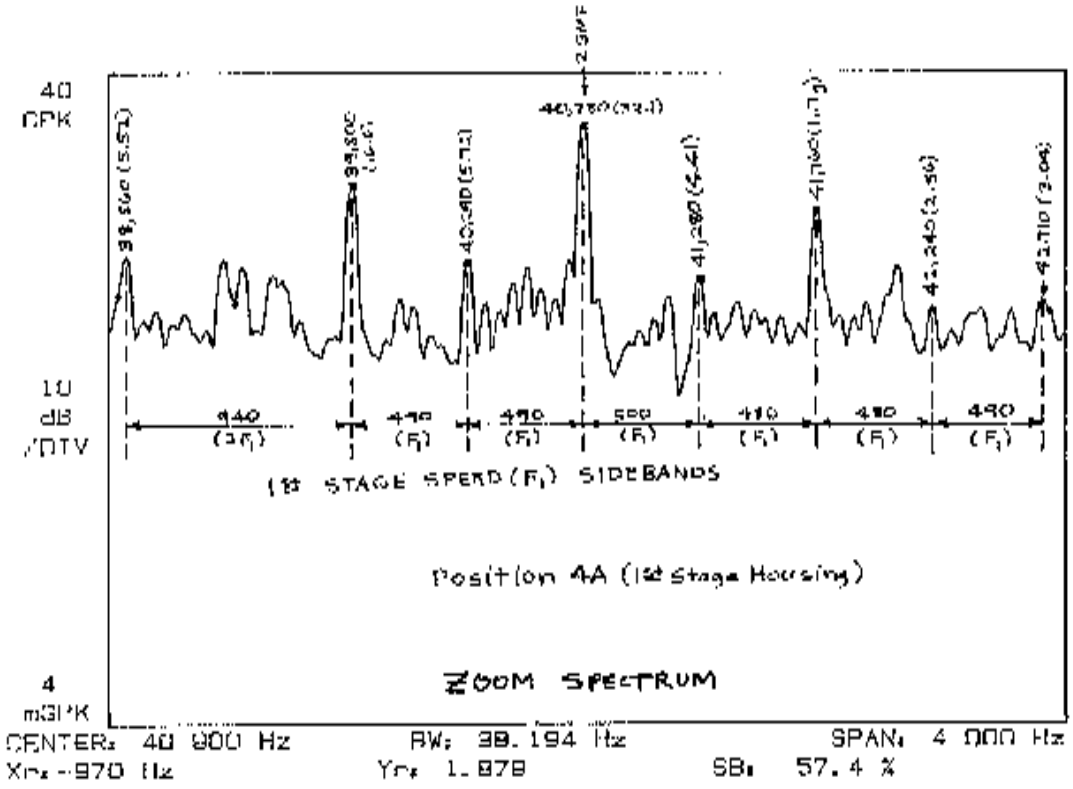


FIGURE 7: FIRST STAGE SPEED SIDEBANDS AROUND 2X GEAR MESH FREQUENCY

© Copyright 2001 Technical Associates of Charlotte, P.C.

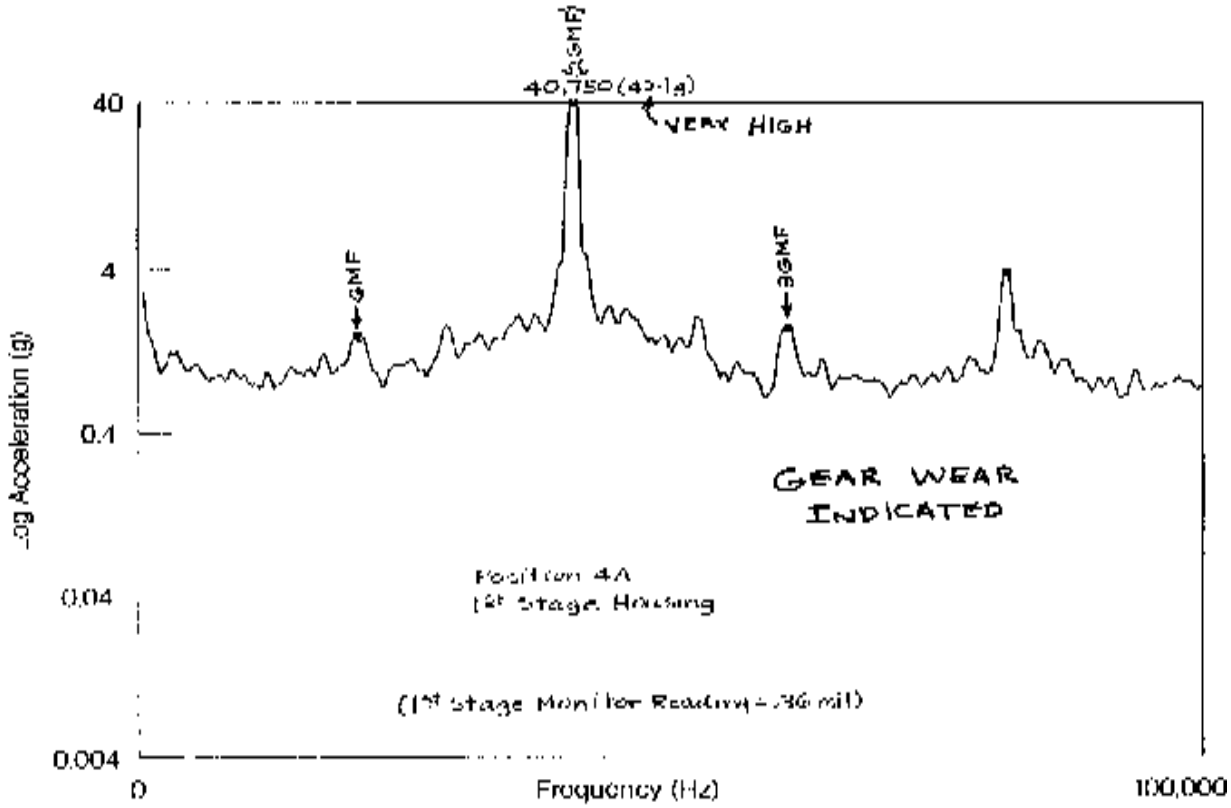


FIGURE 8: HIGH ACCELERATION DUE TO GEAR WEAR 9 MONTHS AFTER BASELINING

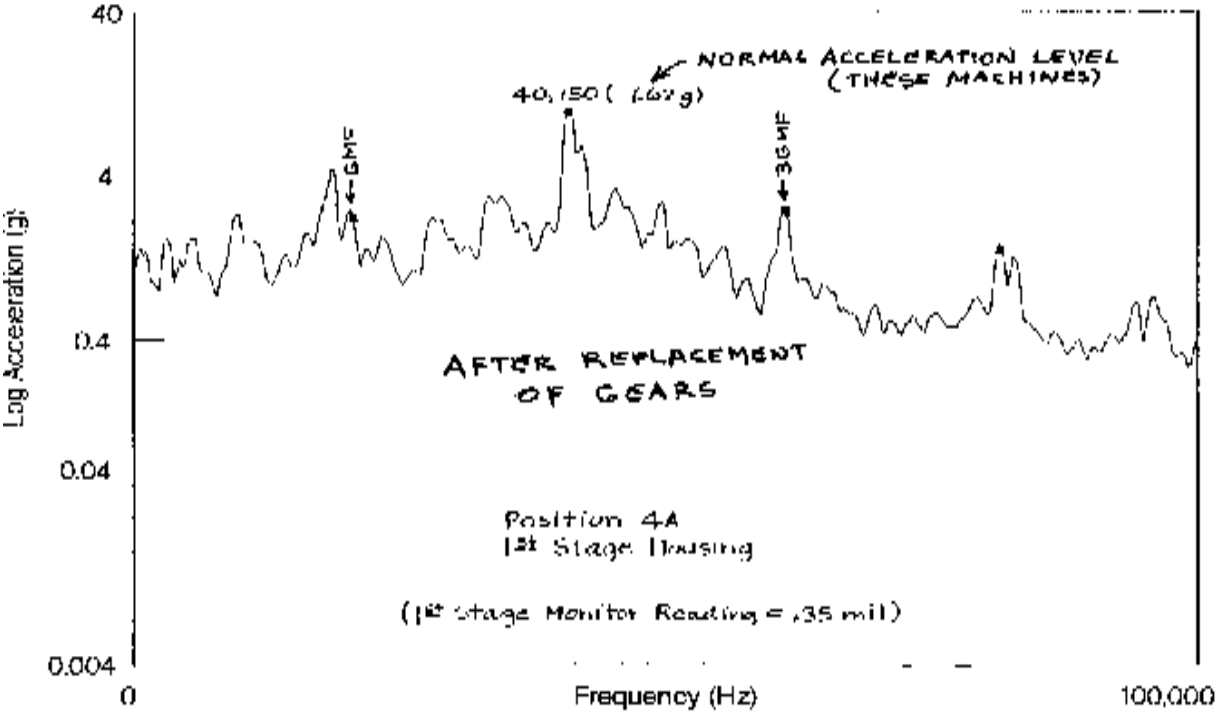


FIGURE 9: SPECTRUM AFTER REPLACEMENT OF BULLGEAR, 1ST, 3RD & 4TH STAGE PINIONS

© Copyright 2001 Technical Associates of Charlotte, P.C.

SEVERE VIBRATION TREND OF A TIMKEN ROLLER BEARING INSTALLED IN A CIRCULATING FAN

I. BACKGROUND

F 21 Circulating Fan #4 suffered a severe bearing frequency vibration problem with its inboard fan bearing which is pictured as Position 3 Figure 1. Note that Timken model 34478 bearings are installed on the fan which is overhung from these two bearings. Severe vibration and spike energy levels were measured during the August 3, 1990 survey which mandated close analysis to prevent potentially catastrophic failure. This machine is analyzed on a regularly scheduled basis as part of a Predictive Maintenance Program annual contract.

II. ANALYSIS RESULTS

Figure 2 shows the vibration spectrum measured on Position 3A of the F 21 Circulating Fan #4 taken on August 3. Note that both Bands 5 and 6 were in alarm especially due to frequencies at 12,870 CPM (Band 5) and 25,740 CPM (Band 6). Comparison with the Timken 34478 bearing defect frequencies in Figure 1 confirmed that the frequency at 12,870 CPM was the outer race frequency (BPFO) while that at 25,740 CPM was its second harmonic (2X BPFO).

Figure 3 shows the trend of spike energy and that of the Band 6 velocity level between March 30, 1989 and October 11, 1990. Note the tremendous increase in spike energy from .630 to 2.89 g/SE on August 3. Figure 4 shows two monthly measurement reports which displayed overall vibration at each point both before bearing replacement (8-3-90 report) and after bearing replacement (10-11-90 report). Note the significant decrease in both vibration and spike energy for fan Positions 3A, 3H and 3V which occurred on the October 11 report (decreased from 80% to 93%) after replacement of the fan bearings.

Figure 5 shows a waterfall plot which illustrates how the vibration spectrum at Position 3A changed from March 30, 1989 through October 11, 1990. Note that no bearing frequencies (or any high frequency energy for that matter), was present on either the March 30 or April 25 signatures. However, high amplitude outer race frequencies did appear as of the December 12 spectrum (.270 in/sec at 2X BPFO). Then, both BPFO and 2X BPFO roughly doubled as of the January 18, 1990 spectrum. They then continued to increase with the next spectrum taken August 3 up to the final levels of .150 in/sec at BPFO and an excessive .775 in/sec at 2X BPFO.

III. RECOMMENDATIONS MADE

Technical Associates recommended immediate replacement of the fan bearings in order to prevent potential catastrophic failure (recommendations have been made to replace these bearings since the January 18 spectrum was captured). In addition, it was recommended that the plant take steps to balance the fan which had trended from only about .120 in/sec at fan speed up to approximately .320 in/sec as of the August 3 spectrum.

© Copyright 2001 Technical Associates of Charlotte, P.C.

IV. CORRECTIVE ACTIONS TAKEN

The plant did follow through with replacement of each of the fan bearings, but did not schedule the fan balancing at this time. Also, the plant took extra efforts to ensure good alignment between the motor and fan shafts.

V. RESULTS OF CORRECTIVE ACTIONS

Figure 5 shows that all higher frequencies were eliminated in the spectrum of October 11 taken after bearing replacement. In addition, some improvement in fan vibration occurred even without the fan balance. Similarly, Figure 3 shows that both the spike energy and the overall velocity levels dropped significantly following bearing replacement.

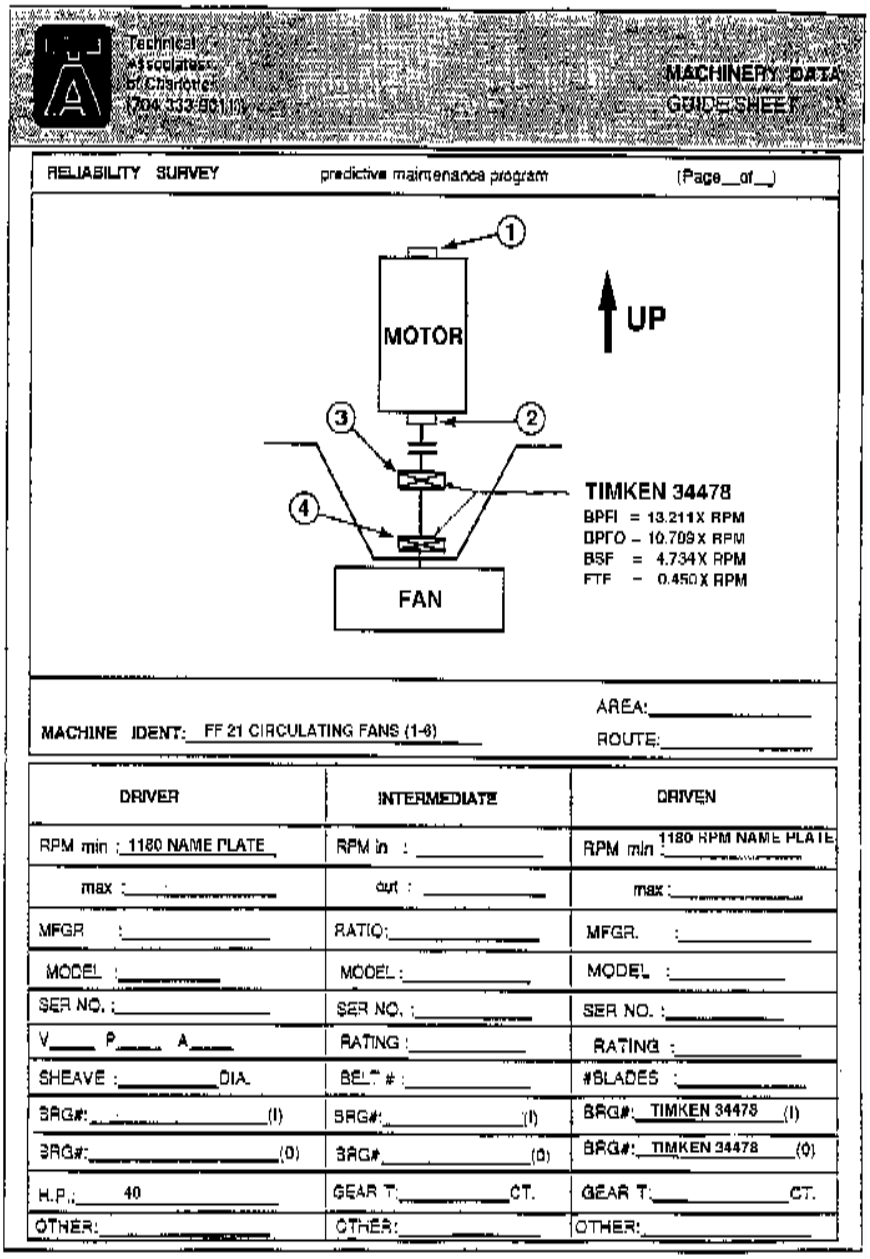


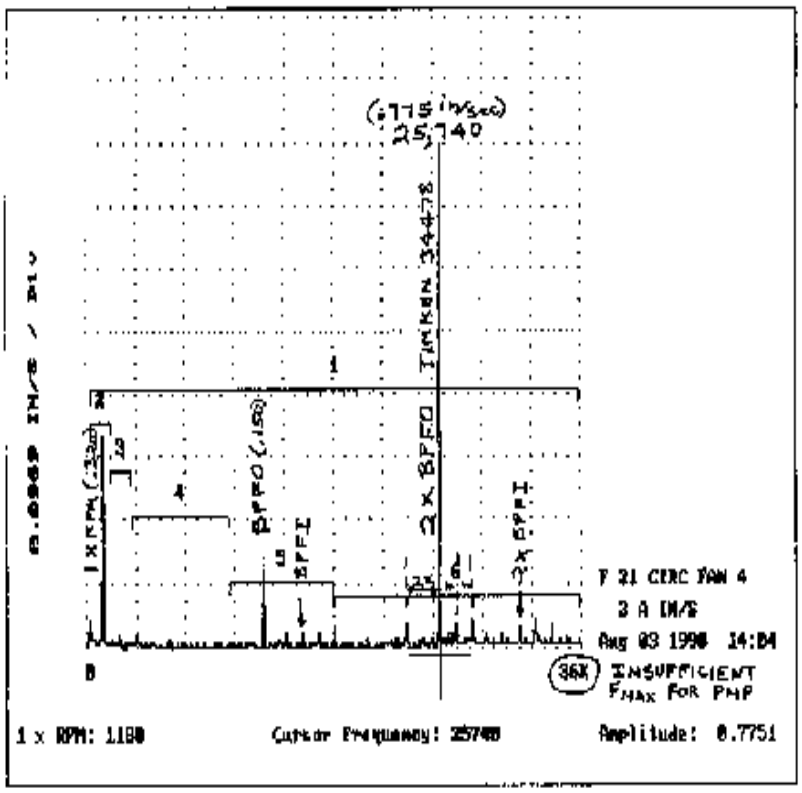
FIGURE 1

DCB-8

CF5

R-9210-1

© Copyright 2001 Technical Associates of Charlotte, P.C.



002% x 1.000 = 0.0200 1 x RPM: 1180.00

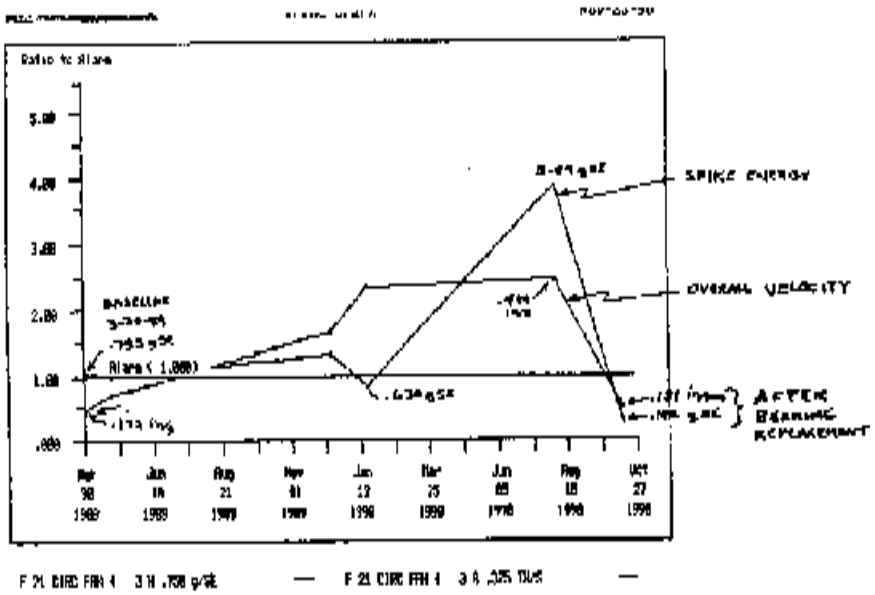
Machine	Pos	Dir	Units	Date
F 21 CIRC FAN 4	3	A	IN/S	Aug 03 1998 14:04

Frequency	1 x RPM	Amplitude	Frequency	1 x RPM	Amplitude
180	0.153	0.0450	31500	26.695	0.0300←2BPFI
1170	0.992	0.3199	32670	27.686	0.0450
2340	1.983	0.0200	33840	28.678	0.0349←2BPFI + 2X
3800	3.051	0.0200	35010	29.669	0.0249
12870	10.907	0.1500←BPFO			
14580	12.356	0.0200			
15750	13.347	0.0249←BPFI			
16920	14.339	0.0249			
18090	15.331	0.0300			
23400	19.831	0.0349← 2BPFO - 2X RPM			
25740	21.814	0.7751← 2X BPFO			
26370	22.347	0.0249			
26730	22.653	0.0249			
26910	22.805	0.0349← 2 BPFO + 1X RPM			
28080	23.797	0.0450← 2 BPFO + 2X RPM			
29070	24.636	0.0249← 2 BPFI - 2X RPM			
30330	25.703	0.0200← 2 BPFI - 1X RPM			

T - Threshold adjust Q - Quit

FIGURE 2

© Copyright 2001 Technical Associates of Charlotte, P.C.



IRD Mechanalysis Data Analysis System

Machine F 21 CIRC FAN 4 Pos 3 Dir H Units g/SE Alarm .750

Ampl	Insp Code	Date mm/dd/yy	Time	Ampl	Insp Code	Date mm/dd/yy	Time
.755	0	03-30-89	10:57				
.744	0	04-25-89	14:51				
.979	0	12-12-89	11:49				
.630	0	01-18-90	11:33				
2.89	0	08-03-90	14:05				
.182	0	10-11-90	15:07				

IRD Mechanalysis Data Analysis System

Machine F 21 CIRC FAN 4 Pos 3 Dir A Units IN/S Alarm .375

Ampl	Insp Code	Date mm/dd/yy	Time	Ampl	Insp Code	Date mm/dd/yy	Time
.172	0	03-30-89	10:57				
.263	0	04-25-89	14:51				
.628	0	12-12-89	11:49				
.872	0	01-18-90	11:33				
.919	0	08-03-90	14:04				
.181	0	10-11-90	15:07				

FIGURE 3

© Copyright 2001 Technical Associates of Charlotte, P.C.

IRD Mechanalysis Data Analysis System
Nov-23-90

Machine Report (Latest Data) for: F 21 CIRC FAN 4

Date	Time	Pos	Dir	Ampl	Units	Change	Alarm	Inspection
Oct-11-90	15:05	1	A	.285	IN/S	21%	.375	
Oct-11-90	15:05	1	H	*.538	IN/S	267%	.375	
Oct-11-90	15:05	1	H	.012	g/SE	-94%	.500	
Oct-11-90	15:05	1	V	*.466	IN/S	134%	.375	
Oct-11-90	15:05	2	A	.096	IN/S	-64%	.375	
Oct-11-90	15:06	2	H	.182	IN/S	31%	.375	
Oct-11-90	15:06	2	H	.026	g/SE	-81%	.500	
Oct-11-90	15:06	2	V	.275	IN/S	-6%	.375	
Oct-11-90	15:07	3	A	.181	IN/S	-80%	.375	
Oct-11-90	15:07	3	H	.039	IN/S	-91%	.375	
Oct-11-90	15:07	3	H	.192	g/SE	-93%	.500	
Oct-11-90	15:07	3	V	.077	IN/S	-80%	.375	

12 lines this report

10-11-90 MEASUREMENT REPORT

Hit any key to continue

IRD Mechanalysis Data Analysis System Page 1
Nov-23-90

Machine Report (Latest Data) for: F 21 CIRC FAN 4

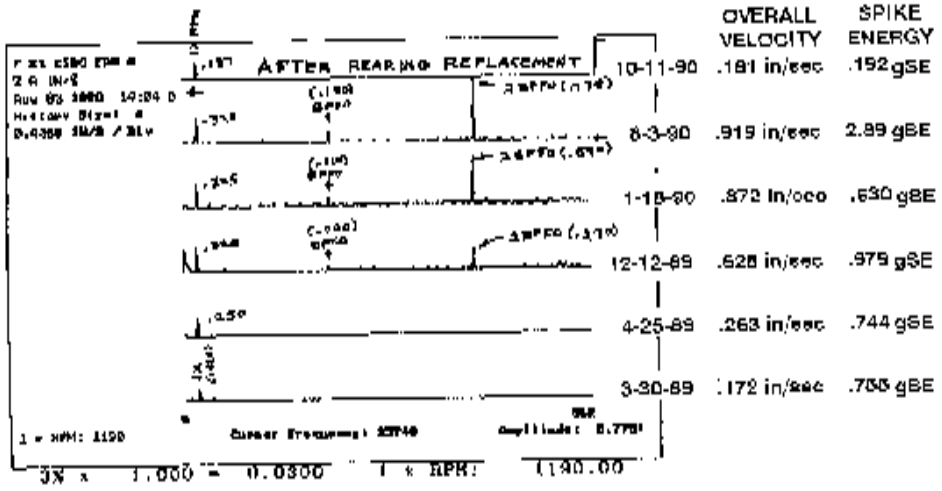
Date	Time	Pos	Dir	Ampl	Units	Change	Alarm	Inspection
Aug-03-90	14:24	1	A	.236	IN/S	36%	.375	
Aug-03-90	14:24	1	H	.146	IN/S	-51%	.375	
Aug-03-90	14:24	1	H	.207	g/SE	527%	.500	
Aug-03-90	14:24	1	V	.189	IN/S	2%	.375	
Aug-03-90	14:24	2	A	.266	IN/S	29%	.375	
Aug-03-90	14:24	2	H	.124	IN/S	33%	.375	
Aug-03-90	14:24	2	H	.149	g/SE	51%	.500	
Aug-03-90	14:25	2	V	.293	IN/S	17%	.375	
Aug-03-90	14:04	3	A	*.919	IN/S	5%	.375	
Aug-03-90	14:05	3	H	*.452	IN/S	12%	.375	
Aug-03-90	14:05	3	H	*2.89	g/SE	776%	.500	
Aug-03-90	14:05	3	V	*.383	IN/S	22%	.375	
Aug-03-90	14:06	4	A	*.380	IN/S	-16%	.375	
Aug-03-90	14:06	4	H	*.627	IN/S	44%	.375	
Aug-03-90	14:06	4	H	*2.94	g/SE	167%	.500	
Aug-03-90	14:06	4	V	*.625	IN/S	101%	.375	

16 lines this report

8-3-90 MEASUREMENT REPORT

FIGURE 4

© Copyright 2001 Technical Associates of Charlotte, P.C.



Machine	Pos	Dir	Units	Date
F 21 CIRC FAN 4	3	A	IN/S	Aug 09 1990 14:04

Frequency	1 x RPM	Amplitude	
180	0.151	0.0450	
1170	0.983	0.3198	← 1X RPM
12870	10.815	0.1500	← BPFO
18080	15.202	0.0300	
23400	18.654	0.0342	-9X RPM
25740	21.630	0.7751	← 2BPFO
26910	22.613	0.0349	+1X RPM
28080	23.587	0.0450	12X RPM
31500	26.471	0.0300	
32670	27.454	0.0450	
33840	28.437	0.0348	

004% x 1.000 = 0.0400 1 x RPM: 1190.00

Machine	Pos	Dir	Units	Date
F 21 CIRC FAN 4	3	A	IN/S	Jan 18 1990 11:39

Frequency	1 x RPM	Amplitude	
1170	0.983	0.3250	← 1X RPM
2340	1.968	0.0400	
12870	10.815	0.1098	← BPFO
25650	21.555	0.5800	← 2BPFO
26820	22.538	0.0850	← +1X RPM
31320	26.319	0.0400	
32580	27.378	0.0750	
33840	28.437	0.0850	
34380	28.891	0.0400	
34920	29.345	0.0800	
35190	29.571	0.0400	

003% x 1.000 = 0.0300 1 x RPM: 1190.00

Machine	Pos	Dir	Units	Date
F 21 CIRC FAN 4	3	A	IN/S	Dec 12 1989 11:49

Frequency	1 x RPM	Amplitude	
90	0.076	0.2850	
300	0.303	0.0950	
1170	0.983	0.3250	← 1X RPM
12870	10.815	0.0600	← BPFO
15750	13.235	0.0400	
25740	21.630	0.3701	← 2BPFO
26910	22.613	0.0300	
28070	24.428	0.0300	
30240	25.412	0.0300	
31500	26.471	0.0300	
33300	27.983	0.0400	
33840	28.437	0.0400	
34390	28.901	0.0300	

FIGURE 5

© Copyright 2001 Technical Associates of Charlotte, P.C.

DETECTION OF SERIOUS BEARING PROBLEMS WITHIN A 400 HP MOTOR SERVING A HOFFMAN BLOWER

I. BACKGROUND

On January 5, 1988, a client requested the services of Technical Associates to evaluate a potential problem on its 400 HP motor which served a 2-Stage Hoffman blower which was a very critical machine. Measurements by the client with his IRD 880 swept filter analyzer showed a significant increase in vibration, particularly in the higher frequencies of the spectrum. However, due to the wide frequency bandwidth of the IRD 880, it was not possible to separate the bearing frequencies from running speed harmonics. Therefore, the services of Technical Associates using a real-time analyzer were requested to confirm the source of the problem and to make recommendations to correct any problems detected.

II. ANALYSIS RESULTS

Figure 1 shows where measurements were made on the motor and blower bearing housings. Note that each of the motor bearings were Fafnir model 7209 bearings which have each of the following defect frequencies:

Inner Race Freq.	=	BPFI	=	8.185X RPM
Outer Race Freq.	=	BPFO	=	5.816X RPM
Ball Spin Freq.	=	BSF	=	2.486X RPM
Cage Freq.	=	FTF	=	0.415X RPM

Plot 1 is a zoom spectrum that showed the motor running speed (F_M) was 3586 RPM. Plot 2 was taken on motor Position 1H which is shown in Figure 1 on the outboard bearing. It showed low vibration at 1X and 2X RPM, each of which was less than .06 in/sec. However, it also showed a .200 in/sec peak at about 29,700 CPM along with frequencies surrounding this peak. Therefore, a zoom spectrum centering on this 29,700 CPM frequency was captured and is shown in Plot 3. This zoom spectrum showed the actual frequency to be about 29,730 CPM which corresponded to 8.19X RPM. Referring back to the bearing defect frequency list, this confirmed an inner race defect frequency.

Also, referring back to Plot 3, note that it was surrounded by peaks at a spacing of about 1385 CPM (.40X RPM). Therefore, this confirmed the presence of sidebands spaced at cage frequency (FTF) around BPFI. Normally, cage frequencies or cage frequency sidebands are one of the last frequencies to show up in a bearing on its path to failure [the normal order of appearance includes the ball pass frequencies (BPFO and BPFI) followed by the ball spin frequency (BSF) and then the cage frequency of both the inner race

© Copyright 2001 Technical Associates of Charlotte, P.C.

and cage frequencies (FTF)]. Thus, the simultaneous appearance caused a good deal of concern on the health of this outboard motor bearing. Plot 5 next shows the comparison of the spectrum on the inboard motor bearing (Plot 5A) with that captured on the outboard motor bearing (Plot 5B). Note the great difference in vibration amplitude at the bearing frequency on each of these two locations (.001 in/sec on the inboard versus .200 in/sec on the outboard). It was concluded that the likely reason for the great difference was considerably more wear on the outboard motor bearing.

III. RECOMMENDATIONS

Because this simultaneous presence of both inner race frequency itself along with sidebands at cage frequency was confirmed, and due to the fact that this machine was so critical, it was recommended that the plant replace each of the motor bearings at first opportunity to prevent potential catastrophic failure of one or both of these bearings. In addition, recommendations were made on alignment specifications which were 2 mils parallel offset maximum and .3 mil/inch of slope maximum angularity. It was recommended that the manufacturer be consulted on the thermal offset required for cold alignment.

DETECTION OF A SERIOUS PROBLEM WITH A HOFFMAN BLOWER BEARING

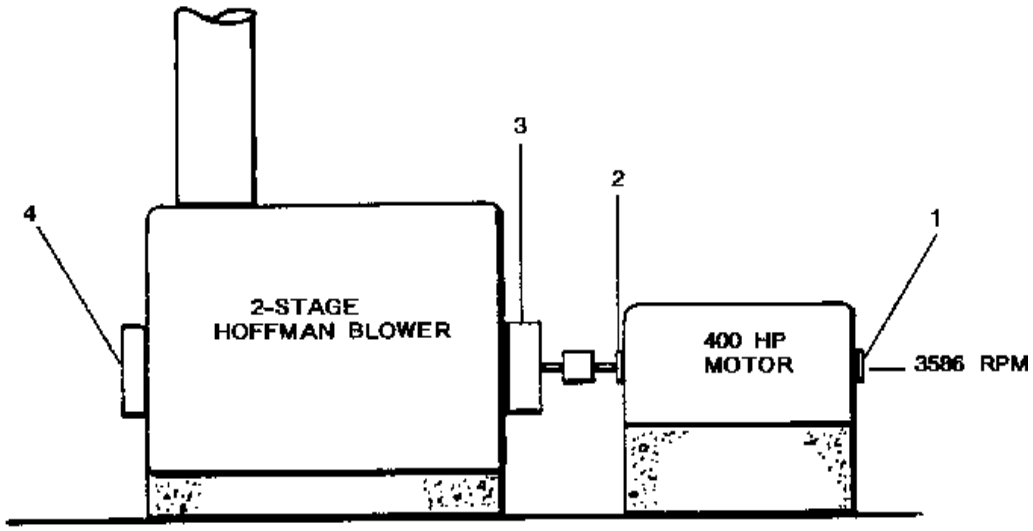
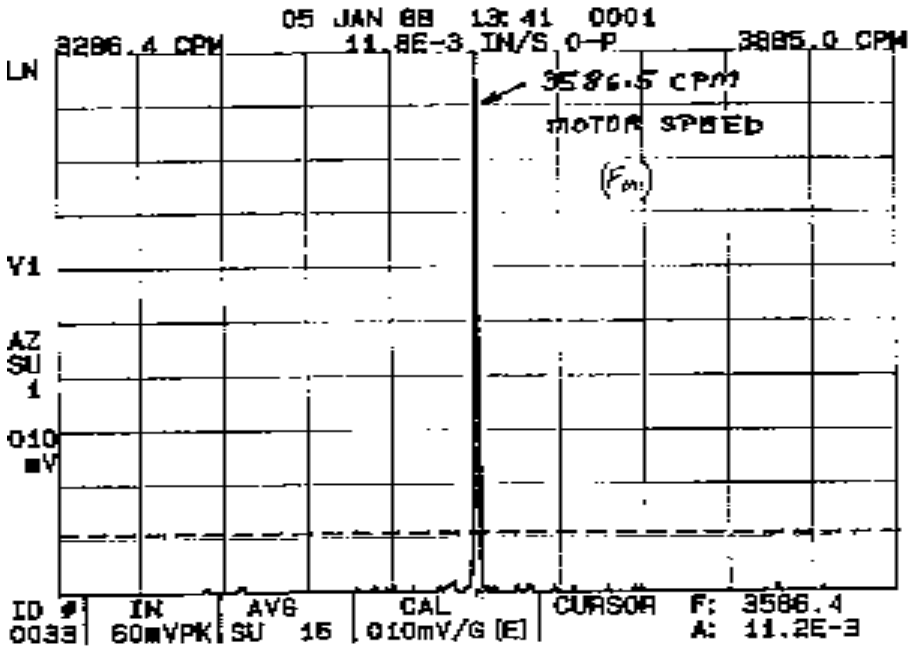


FIGURE 1
HOFFMAN BLOWER
#5
VIBRATION MEASUREMENT LOCATIONS

© Copyright 2001 Technical Associates of Charlotte, P.C.

© Copyright 2001 Technical Associates of Charlotte, P.C.



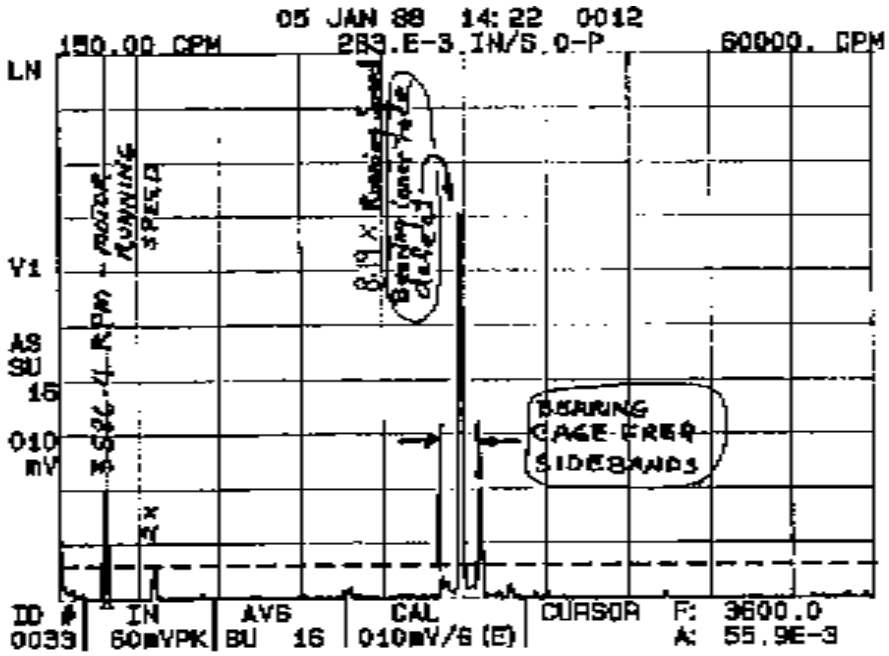
PEAK PICK ON TRACE 1

IN/S 0-P	CPM
11.3E-3	3586.5

06-JAN-88

07: 24: 47

PLOT 1



PEAK PICK ON TRACE 1

IN/S 0-P	CPM
55.6E-3	3600.0
200.E-3	29699.
56.9E-3	31199.

POSITION 1H

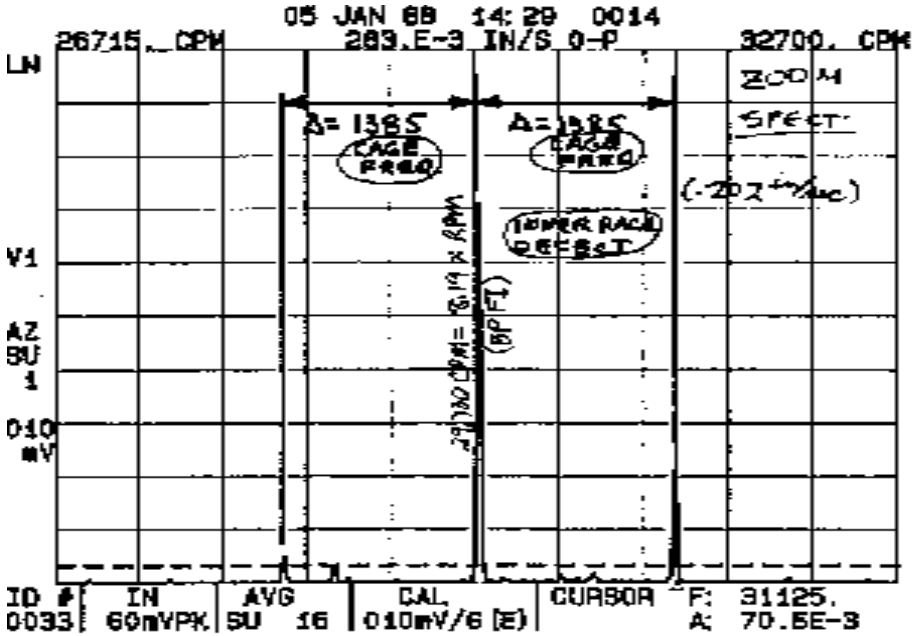
06-JAN-88

07: 43: 53

PLOT 2

© Copyright 2001 Technical Associates of Charlotte, P.C.

© Copyright 2001 Technical Associates of Charlotte, P.C.



PEAK PICK ON TRACE 1

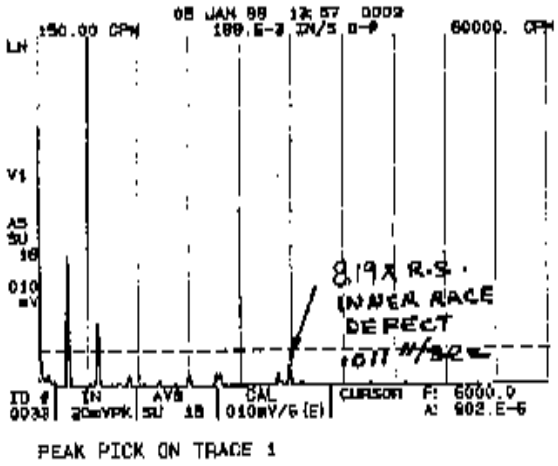
IN/S 0-P	CPM
14.2E-3	28350.
202.E-3	29730.
69.8E-3	31125.

POSITION 1H

06-JAN-88

08:00:52

PLOT 3

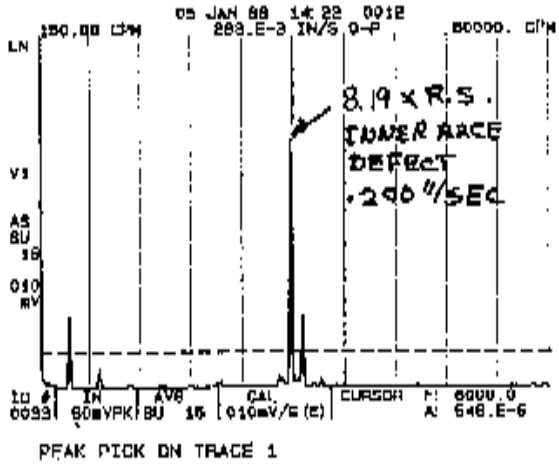


IN/S 0-P	CPM
70.8E-3	3600.0
34.5E-3	7200.0

PLOT 5A

MOTOR INBOARD BEARING
AXIAL MEASUREMENT
POS. 2A

06-JAN-88



IN/S 0-P	CPM
55.6E-3	3600.0
200.E-3	29699.
56.9E-3	31199.

PLOT 5B

MOTOR OUTBOARD BEARING
AXIAL MEASUREMENT
POS. 1A

GREAT DIFFERENCE IN VIBRATION AT POSITIONS 1A & 2A

06-JAN-88

10: 02: 47

© Copyright 2001 Technical Associates of Charlotte, P.C.

SURPRISING DETECTION OF A ROLLING ELEMENT BEARING PROBLEM ON A DRAW FRAME INPUT PINION BEARING AFTER REPLACEMENT OF A FALK GEARBOX WHICH HAD BEEN THOUGHT TO BE THE SOURCE OF THE PROBLEM

Subtitle:
(How it Pays to Know the Identity of Components Within Your Critical Machinery)

I. BACKGROUND

Technical Associates of Charlotte, Inc. was contacted with regards to performing **Vibration Diagnostic Testing** on a Draw Frame gearbox shown in Figure 1. The problem was described as a bumping noise and vibration that appeared to be occurring at once per revolution of the gearbox output speed. This draw frame includes each of 30 rolls of 18 inch diameter supported on 9 inch diameter shafts as shown in Figure 2. These rolls are gear box driven and are outfitted with high cost bearings. There are several individual drives for various sections of the 30 draw frame rolls including one like that shown in Figure 1. This draw frame is used in production of a high strength fiber material called staple ranging in density from 2,000,000 to 3,000,000 denier. From Figure 2, the tow band applies very high forces on the order of 30,000 lbs on each of the bearings supporting the draw frame rolls as the staple wraps around some rolls greater than 180 degrees. This can put great strain on many of the draw frame components as well as their drives.

The Falk gearbox shown in Figure 1 had been replaced at a cost of over $36,000 which unfortunately had little or no effect on the high vibration problem. Several draw frame roll bearings had also been replaced which likewise showed no improvement. Therefore, the plant requested outside assistance to help identify just what was the source of the problem and what was required to resolve it.

The **Vibration Diagnostic Testing** was performed on June 16, 1989.

The following report contains CONCLUSIONS, RECOMMENDATIONS, and a DISCUSSION of test data obtained.

II. CONCLUSIONS

A. Surprisingly, the problem in this machine had nothing to do with components in either the gearbox nor motor, but instead with the bearing supporting the input pinion of the draw frame itself (see Figure 1). An inner race defect frequency and harmonics for the FAG Model #22326ES bearing was detected. This is the bearing closest to the draw frame input pinion. The amplitudes and frequencies associated with the bearing defect indicated that the bearing was in the final stages of failure.

© Copyright 2001 Technical Associates of Charlotte, P.C.

B. Once the data on bearing model numbers and gear tooth counts was made available, the analysis was straightforward, although the vibration spectra had initially appeared very strange when such information was unknown.

C. Data obtained showed both the gearbox and motor to be in good condition.

D. Due to the inaccessibility of the large draw frame roll bearings mounted deep within the machine, the mechanical condition of these bearings could not be assessed. Recommendations were made by the analyst while on-site how this situation could be remedied by placing permanently mounted accelerometers in key locations which were identified on machine prints given to the analyst. It was important to take the speed of these rolls into account when specifying both hardware and techniques since they normally ranged from only 32 to 104 RPM (see Figure 2).

III. RECOMMENDATIONS

A. Recommendations were made while on-site to replace the Fag 22326ES Draw Frame input pinion bearings as soon as possible to prevent catastrophic failure which may have been imminent as evidenced by vibration spectra.

B. Monitor the gearbox utilizing the client's existing IRD 890/7090 **Predictive Maintenance Program** hardware and software to detect similar problems in the future. Also, complete the research on identification of bearing and gear components within the draw frame as well as its driving gearboxes and motors. This data could prove to be invaluable preventing not only future failures of these expensive components, but also helping to prevent costly expenditures replacing gearboxes and/or motors that actually are in good operating condition.

IV. DISCUSSION

A. Initial Analysis Without Available Information on Motor, Gearbox and Draw Frame Components:

Upon arrival at the plant, information concerning the motor, gearbox and draw frame bearing model numbers and gear tooth counts was not yet available. However, after capturing several interesting, high amplitude spectra like those shown in Plots 1, 2 and 3, it became mandatory that this information be supplied since the sources of most of the peaks could not be identified.

Plot 1 showed three peaks, only one of which could be identified due to the lack of information on components within the motor, gearbox and draw frame. Plot 1 is a zoom spectra (taken on position 1H) which included the motor speed of 1229.3 CPM. It also contained unidentified peaks at 1183.5 CPM and 1318.5 CPM. However, even though these two peaks were unidentified, it was noticed that their difference frequency was 135 CPM which equaled the output speed of the gearbox (1318.5-1183.5 = 135 CPM).

Data captured on the gearbox output shaft at positions 7A and 8A looked very unusual (see Plots 2 and 3). None of the dominant peaks which included frequencies of 1170, 2370 and 3540 CPM could be identified at that

© Copyright 2001 Technical Associates of Charlotte, P.C.

time. These three frequencies could not be related to either fundamentals nor harmonics of either the gearbox input speed (1229 RPM), intermediate speed (660 RPM) nor output speed (133 RPM). Also unusual about Plots 2 and 3 was a whole series of sideband frequencies which were spaced at gearbox output speed surrounding the 1170, 2370 and 3540 CPM peaks (as confirmed by zoom spectra taken around the approximate 1170 and 2370 CPM peaks shown in Plots 4 and 5). Depending on the cause of these spectra generating this unique pattern of frequencies, it was felt that the problem could in fact be quite severe due to amplitudes up to .351 in/sec and sideband levels up to almost .100 in/sec shown in Plots 4 and 5 (bearings, gears, electrical problems?). However, there simply was not sufficient machine component information available.

Therefore, Technical Associates' analyst strongly asserted this data simply must be made available. Despite several arguments claiming this would require an extended time, the client finally dispatched a technician to research this information as quickly as possible.

B. Subsequent Analysis Once Gearbox Component Data Was Supplied:

Although it took approximately 2 hours, the plant finally was able to supply information on the gearbox, but advised no information was available at that time concerning components within either the motor nor draw frame.

The following mechanical frequencies were calculated based on the motor speed of 1229.3 RPM as seen in Plot 1 and the component information supplied (the speed was also confirmed by a strobe light):

Motor Speed		= 1,229.3 RPM
Input Shaft Gear Mesh	1229.3(22T)	= 27,044 CPM
Intermediate Shaft Speed	27,044.6/41T	= 659.62 RPM
Intermediate Pinion Gear Mesh	659.62(20T)	= 13,192.5 CPM
Output Shaft Speed	13,192.4/99T	= 133.25 RPM

GEARBOX BEARINGS

1. Input Shaft Bearings (Timken Nos. 6559 and 6536)

Outer Race (BPFO)	8.23 (1229.3)	= 10,117.14 CPM
Inner Race (BPFI)	10.77 (1229.3)	= 13,239.56 CPM
Ball/Roller (BSF)	7.14 (1229.3)	= 8,777.20 CPM
Cage (FTF)	.43 (1229.3)	= 538.60 CPM

© Copyright 2001 Technical Associates of Charlotte, P.C.

2. Intermediate Shaft Bearings (Timken Nos. HH224335 and HH224310)

Outer Race (BPFO)	7.11 (659.62) = 4,689.90 CPM
Inner Race (BPFI)	9.88 (659.62) = 6,517.0 CPM
Ball/Roller (BSF)	5.87 (659.62) = 3,872.0 CPM
Cage (FTF)	.42 (659.62) = 277.0 CPM

3. Output Shaft Bearings (Timken No. 99100)

Outer Race (BPFO)	10.19 (133.25) = 1,357.80 CPM
Inner Race (BPFI)	12.81 (133.25) = 1,707.0 CPM
Ball/Roller (BSF)	8.40 (133.25) = 1,119.30 CPM
Cage (FTF)	.44 (133.25) = 58.60 CPM

Even after compiling all this data, the dominant peaks in spectra shown in Plots 1 through 5 still could not be pinpointed. Each of these spectra were dominated by a frequency of about 1183 CPM or multiples of this frequency. Therefore, information was again requested on the components within the motor and the draw frame which is driven by this gear box. Several technicians now took approximately 4 hours to research this data. Finally, they were able to acquire information on the draw frame components. This finally uncovered the source of the real problem which was never within gearbox components at all. This will next be discussed.

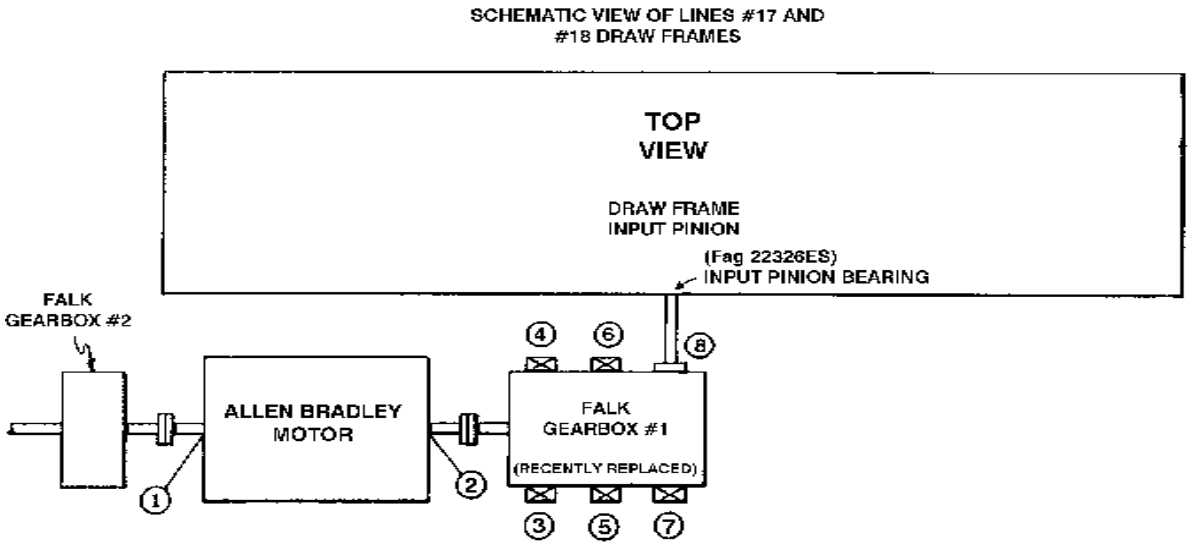


FIGURE 1

DETECTION OF A ROLLING ELEMENT BEARING PROBLEM IN A DRAW FRAME GEARBOX

© Copyright 2001 Technical Associates of Charlotte, P.C.

C. Results After Identification of Motor and Draw Frame Components:

Following below are calculated mechanical frequencies for the draw frame components once this information was finally made available:

DRAW FRAME PINION BEARINGS

1. Inboard Bearing, near Input Pinion (FAG #22326ES)

Outer Race	(BPFO)	6.12 (133.25) =	815.50 CPM	
Inner Race	(BPFI)	8.88 (133.25) =	1,183.3 CPM	**Problem Source**
Ball/Roller	(BSF)	2.65 (133.25) =	353.10 CPM	
Cage	(FTF)	.41 (133.25) =	54.60 CPM	

2. Outboard Bearing (Timken #22226ES)

Outer Race	(BPFO)	8.22 (133.25) =	1,095.30 CPM
Inner Race	(BPFI)	10.80 (133.25) =	1,439.10 CPM
Ball/Roller	(BSF)	3.53 (133.25) =	470.40 CPM
Cage	(FTF)	.43 (133.25) =	57.30 CPM

Once the information shown above was made available, the solution was straightforward. Please refer to Plots 6 and 7 which are repeats of zoom Plots 4 and 5, but with each of the formerly unidentified peaks now pin-pointed. Plots 6 and 7 are zoom spectra with a resolution of 3 CPM. Therefore, the peaks at 1185 and 2367 CPM were confirmed to be 1X BPFI and 2X BPFI for the FAG 22326ES draw frame input pinion bearing, respectively. Then, each of the 135 CPM sidebands surrounding them were found to be spaced at 1X RPM of the draw frame input pinion bearing.

There are normally four (4) stages of rolling element bearing failure as shown in Figure 3. In the early stages of bearing failure, the natural frequencies of bearing components (races, rolling elements, cages, etc.) are excited. These frequencies usually occur between 30,000 and 120,000 CPM. During later stages of failure, bearing defect frequencies appear. Then, as the defect propagates around the periphery of the bearing races, harmonics of defect frequencies appear and develop sidebands normally spaced at running speed when wear becomes clearly visible. In the final stage, 1X RPM will increase in amplitude and the defect frequencies will diminish or disappear completely. At this point, the bearing can fail at any time.

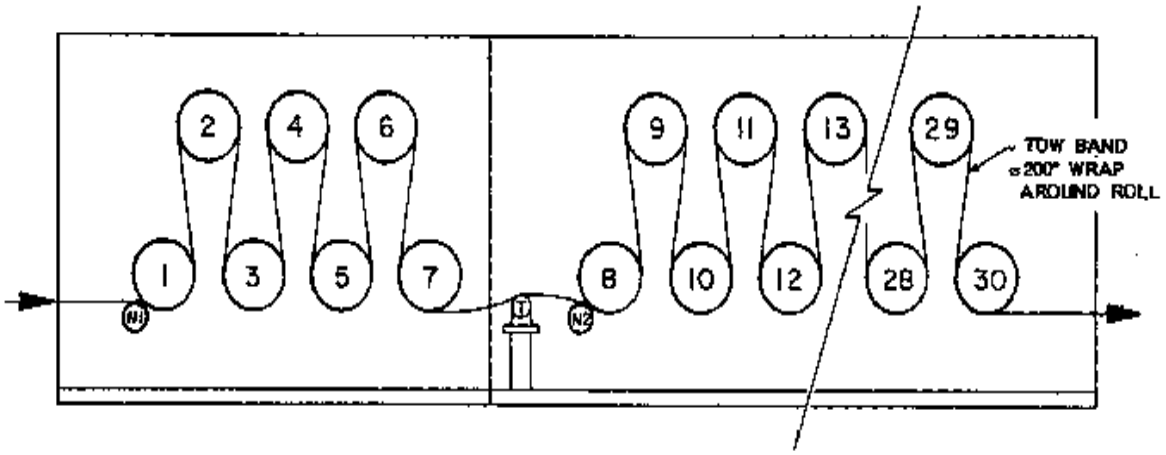
Since the bearing wear was obviously well into late stage 3, it was strongly felt that wear of this bearing was likely quite severe. This was particularly concerning when it was realized that the spectrum of Plot 1 was captured all the way back on position 1H on the motor outboard bearing; even here the dominant vibration was not at 1X motor RPM, but

© Copyright 2001 Technical Associates of Charlotte, P.C.

instead at 1X BPFI of the draw frame pinion bearing remotely located from this position (see Figure 1). Therefore, the analyst recommended replacing the bearing as soon as possible.

V. RESULTS OF CORRECTIVE ACTIONS

NOTE: Plot 8A was taken on the same machine at Gearbox Position 8A after replacement of the draw frame input pinion bearing (FAG 22326ES). It compares directly to Plot 3 taken before (which is shown below Plot 8A as "Plot 8B" to help the reader in spectral comparison). Note the significant drop in vibration and the total elimination of draw frame bearing defect frequencies. As of the writing of this case history almost 2 years since this diagnostic investigation, the draw frame has not experienced any further failures and continues to enjoy low vibration and noise levels.



ROLL INDENT	490 FPM (RPM)	150 FPM (RPM)	ROLL DIAMETER
1 - 5	31.7	10.2	18.0"
6	67.6	21.8	18.0"
7	94.6	30.5	18.0"
8 - 30	104.0	33.5	18.0"
NIP-1	71.3	23.0	8.0"
NIP-2	234.0	75.4	8.0"
TENSION	651.0	210.0	2.875"

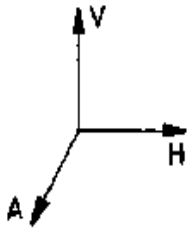
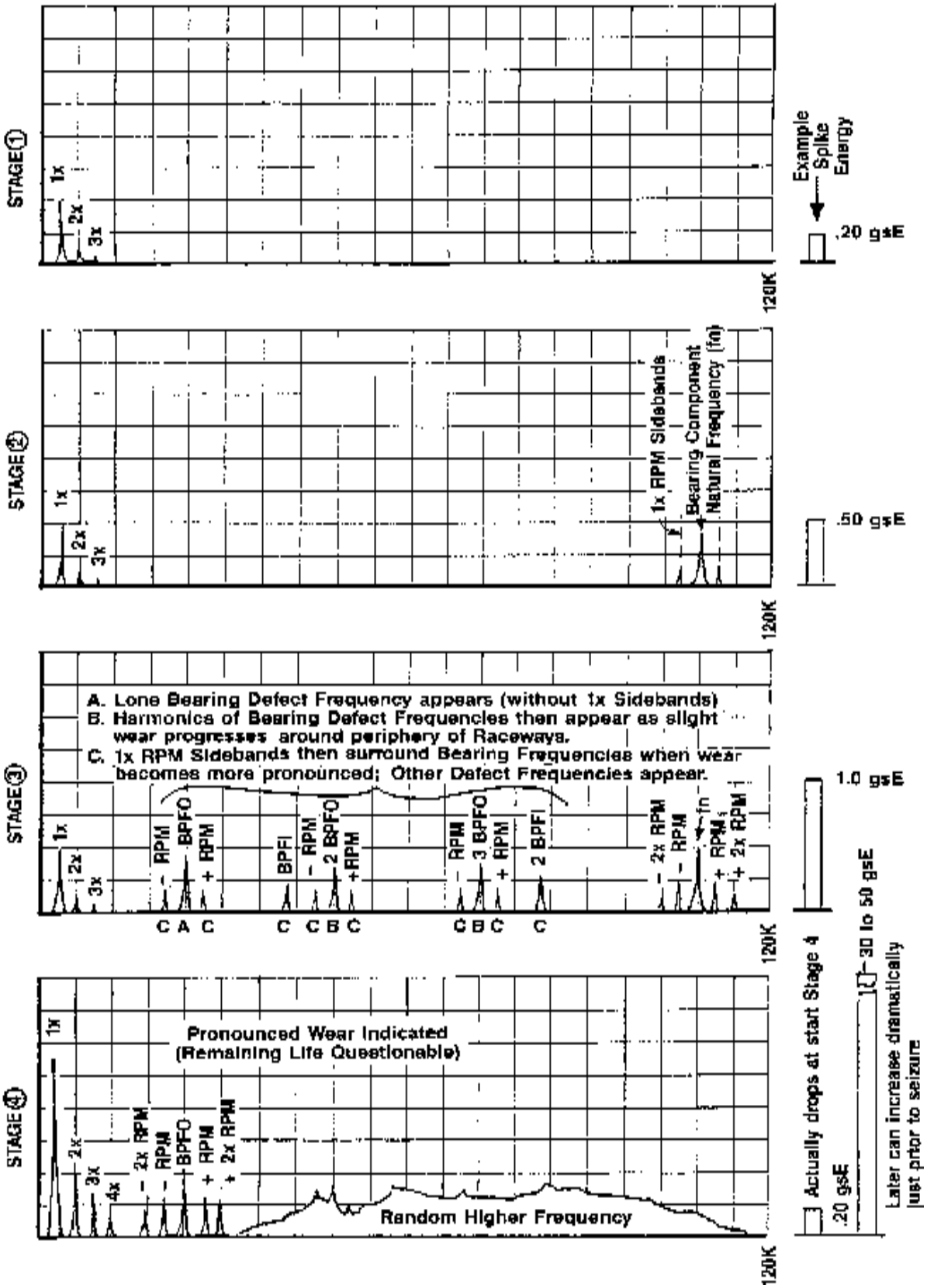


FIGURE 2
TENSION METER SET-UP ON DRAW FRAME

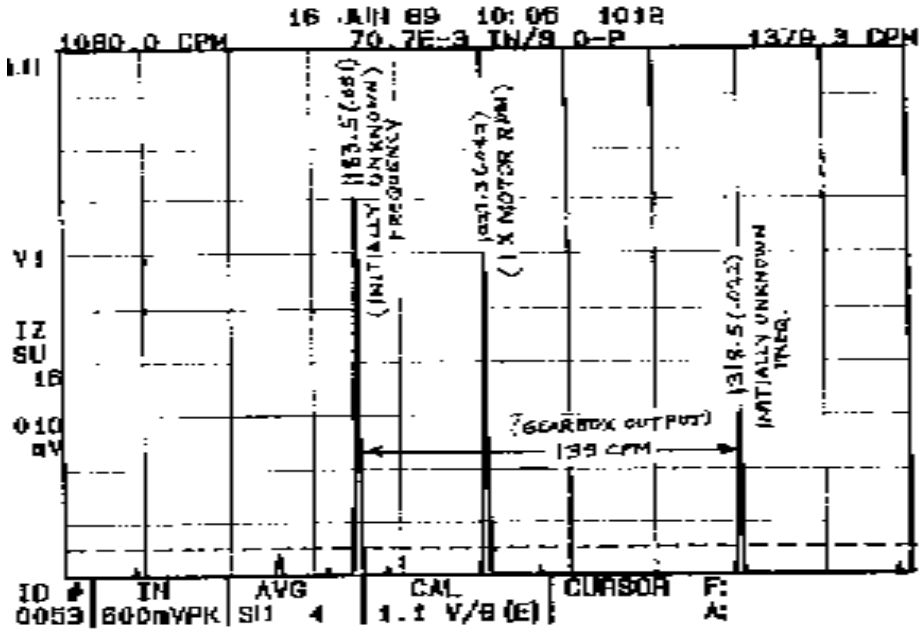
DF1 R-9210-1

© Copyright 2001 Technical Associates of Charlotte, P.C.

FIGURE 3. 4 PRIMARY FAILURE STAGES THRU WHICH MOST ROLLING ELEMENT BEARINGS PASS (VELOCITY SPECTRA)



© Copyright 2001 Technical Associates of Charlotte, P.C.



PEAK PICK ON TRACE 1

POSITION 1H

IN/S 0-P	CPM
51.3E-3	1183.5
42.3E-3	1229.3
21.6E-3	1318.5

21-JUN-89

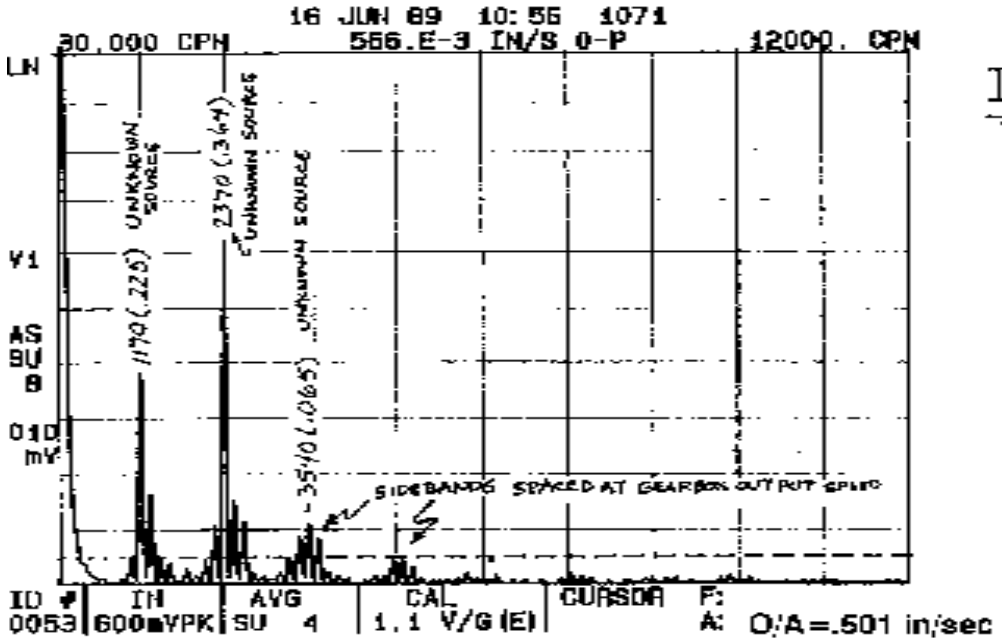
07:28:04

PLOT 1

LINE 18

© Copyright 2001 Technical Associates of Charlotte, P.C.

© Copyright 2001 Technical Associates of Charlotte, P.C.



PEAK PICK ON TRACE 1

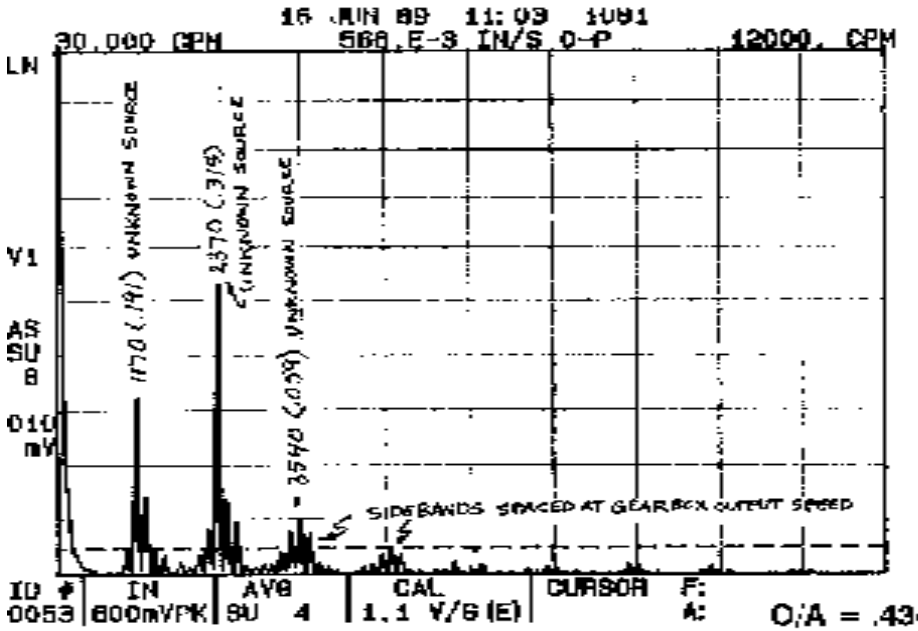
IN/S 0-P	CPM
31.0E-3	1050.0
* 225.E-3	1170.0
98.2E-3	1320.0
31.0E-3	1440.0
62.0E-3	2220.0
* 364.E-3	2370.0
90.4E-3	2490.0
67.2E-3	2640.0
28.4E-3	3269.9
51.7E-3	3420.0
64.6E-3	3540.0
49.1E-3	3690.0

21-JUN-89

07:52:41

PLOT 2

LINE 18



IN/S 0-P	CPM
*191.E-3	1170.0
85.3E-3	1320.0
49.1E-3	2220.0
*315.E-3	2370.0
82.7E-3	2490.0
56.9E-3	2640.0
46.5E-3	3420.0
59.4E-3	3540.0
43.9E-3	3690.0

O/A = .434 in/sec
POSITION 8A

PEAK PICK ON TRACE 1

21-JUN-89

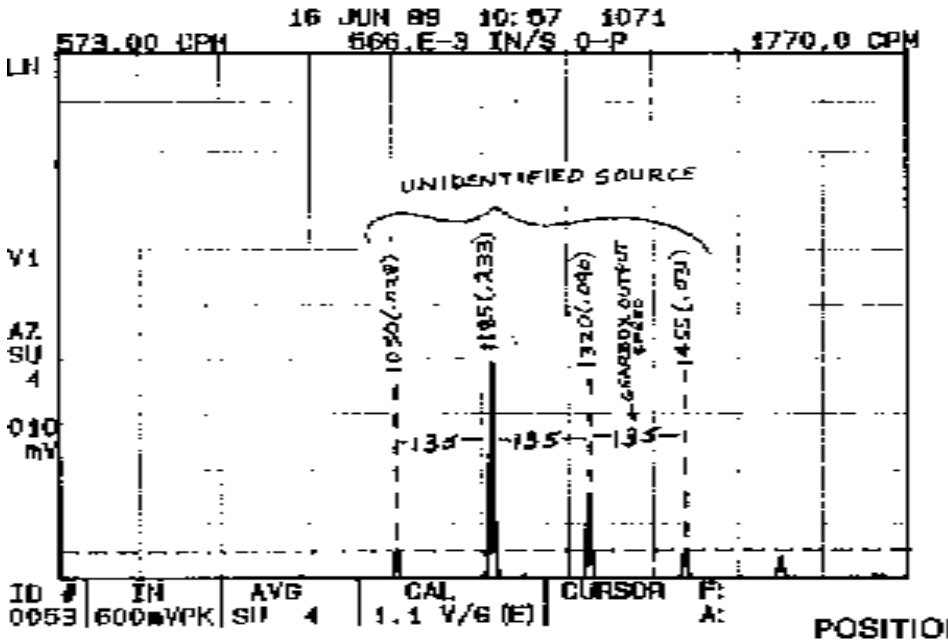
08: 00: 03

PLOT 3

LINE 18

© Copyright 2001 Technical Associates of Charlotte, P.C.

© Copyright 2001 Technical Associates of Charlotte, P.C.



POSITION 7A

PEAK PICK ON TRACE 1

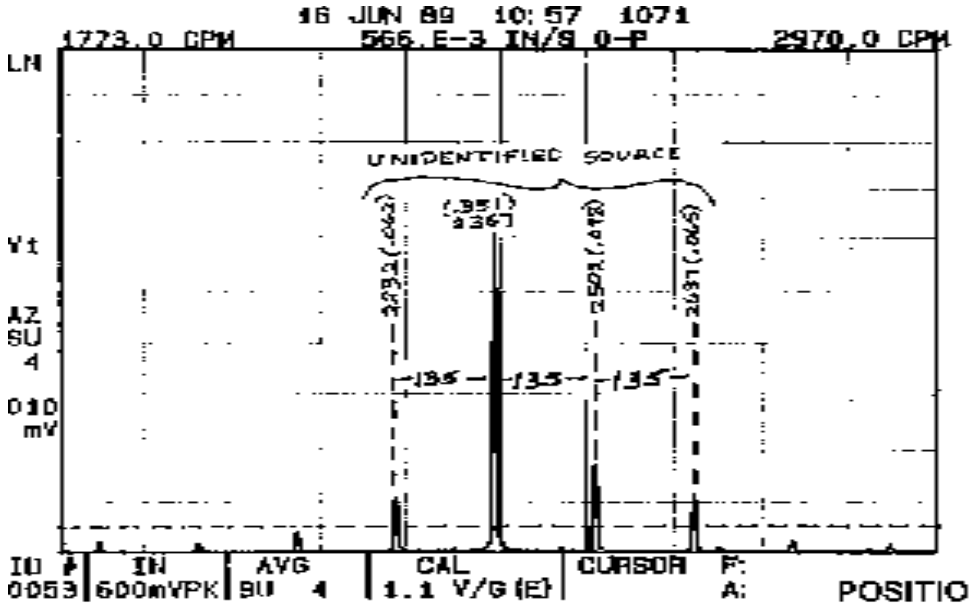
IN/S 0-P	CPM
28.4E-3	1050.0
233.E-3	1185.0
90.4E-3	1320.0
31.0E-3	1455.0

21-JUN-89

07: 54: 43

PLOT 4

LINE 18



POSITION 7A

IN/S 0-P	CPM
62.0E-3	2232.0
351.E-3	2367.0
98.2E-3	2502.0
64.6E-3	2637.0

PEAK PICK ON TRACE 1

21-JUN-89

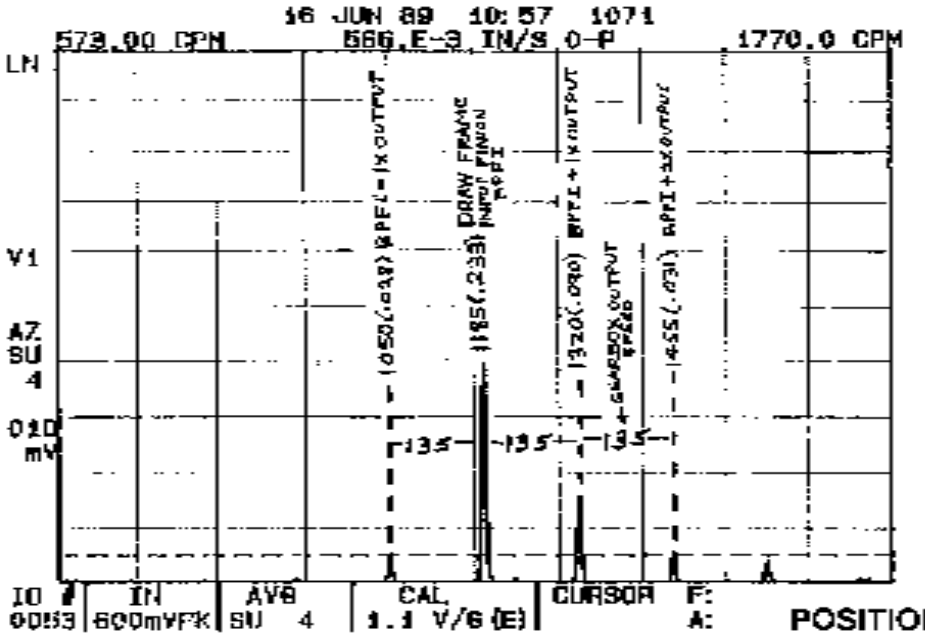
07: 56: 19

PLOT 5

LINE 18

© Copyright 2001 Technical Associates of Charlotte, P.C.

© Copyright 2001 Technical Associates of Charlotte, P.C.



PEAK PICK ON TRACE 1

IN/S 0-P	CPM
28.4E-3	1050.0
233.E-3	1185.0
90.4E-3	1320.0
31.0E-3	1455.0

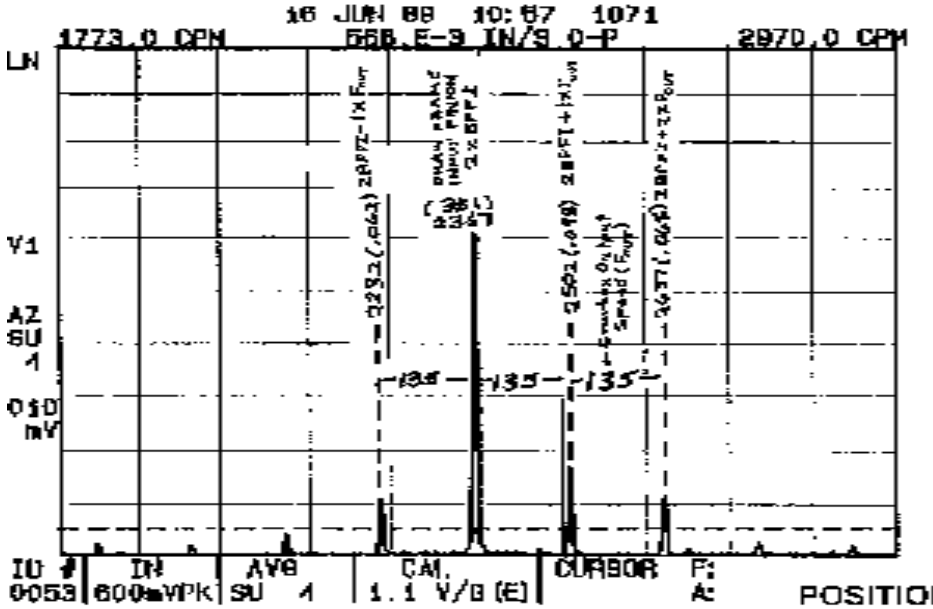
21-JUN-89

07:54:43

SIGNIFICANT WEAR INDICATED OF FAG 22326ES DRAW FRAME INPUT PINION BEARING

PLOT 6

LINE 18



POSITION 7A

PEAK PICK ON TRACE 1

IN/S 0-P	CPM
62.0E-3	2232.0
351.E-3	2367.0
98.2E-3	2502.0
64.6E-3	2637.0

21-JUN-89

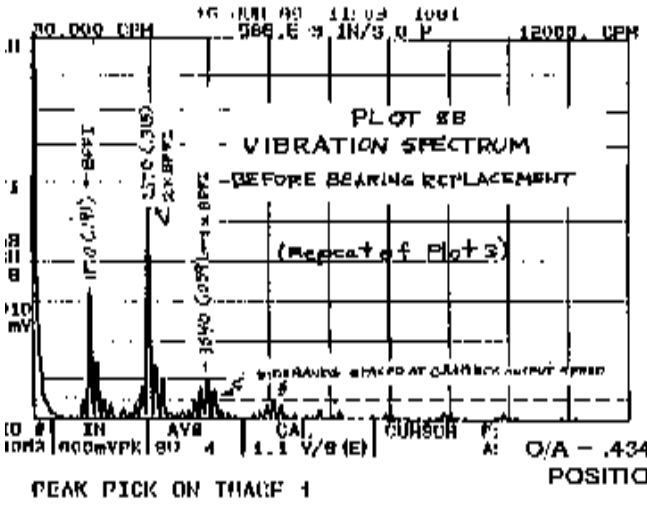
07:56:19

SIGNIFICANT WEAR INDICATED OF FAG 22326ES DRAW FRAME INPUT PINION BEARING

PLOT 7

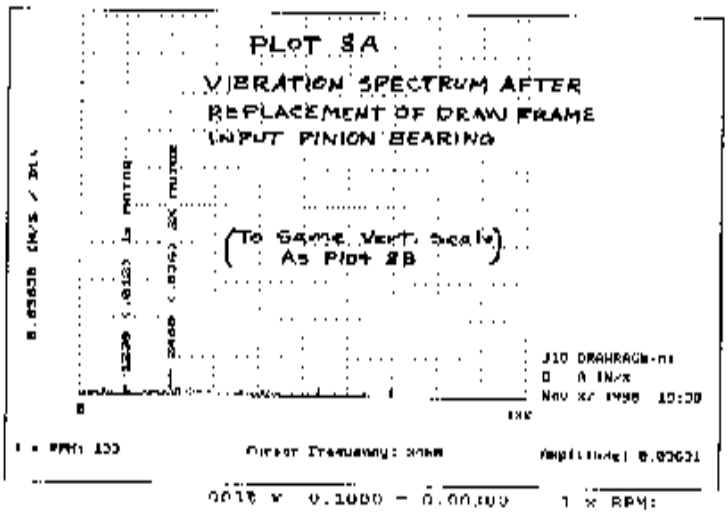
LINE 1B

© Copyright 2001 Technical Associates of Charlotte, P.C.



IN/S 0 P	CPM
*191.E-3	1170.0
85.3E-3	1320.0
49.1E-3	2220.0
*315.E-3	2370.0
82.7E-3	2490.0
56.9E-3	2640.0
46.5E-3	3420.0
59.4E-3	3540.0
43.9E-3	3690.0

21 JUN 89



Machine: 718 DRAWFRAME MT | Proc: B | Dir: A | Units: IN/S | Date: Nov 27 1990 13:30

Frequency	1 x RPM	Amplitude	Frequency	1 x RPM	Amplitude
190	-	0.00516	3520	-	0.00718
480	-	0.00405	3600		0.00456
660	-	0.00509	3690	-	0.00521
1330		0.01241	4230	-	0.00439
1740	-	0.00359	4350	-	0.00589
1980	-	0.00333	4770	-	0.00370
2310	-	0.00451	4920		0.00612
2460	-	0.03631	5100	-	0.00751
2670	-	0.00797	5340	-	0.00492
2790	-	0.00443	6150	-	0.00334
2910	-	0.00786	6420	-	0.00402
3000	-	0.01072	6570		0.01001
3180	-	0.00720			
3240	-	0.00800			
3330	-	0.00446			
3390	-	0.00428			
3450	-	0.00656			

T - Threshold adjust Q - Quit

© Copyright 2001 Technical Associates of Charlotte, P.C.

SERIOUS PROBLEM OF A BEARING TURNING ON A SHAFT OF ONE MOTOR AND DETECTION OF SEVERE BEARING PROBLEMS ON ANOTHER MOTOR AS EVIDENCED BY NUMEROUS OUTER RACE SUM AND DIFFERENCE FREQUENCIES

I. BACKGROUND

On Saturday March 12, it is understood that the Rolling Mill #4 Main Drive Motor had high vibration and noise amplitudes which could even be felt in the floor and heard in other areas. Technical Associates was contacted by the plant on March 15 and brought in to diagnose the problem. By this time, the noise and vibration had dropped significantly, but there was a peculiar noise pulse present each time the machine was started up under load, particularly at lower speeds up to approximately 300 to 400 RPM. Measurements were made on the #4 Main Drive Motor using a real-time analyzer. In addition, an analysis was conducted on the #4 Rewind Inboard Motor which was experiencing high vibration. Following is a Discussion of the measurements that were made on March 15. Measurements were made with a Rockland 5815A narrowband real-time analyzer.

Later, after corrective actions were completed, a Predictive Maintenance Program was instituted at the client's plant by Technical Associates. This allowed capture of vibration spectra following the corrective actions using an IRD 890 and IRD 7090 software.

II. CONCLUSIONS

A. MILL 4 MAIN DRIVE MOTOR (See Figure 1)

Vibration spectra indicate the following:

1. No vibration spectra on this machine exceeded .300 in/sec alarm levels on this March 15 test date at any of the six (6) positions evaluated on the outboard and inboard motor bearings (See Plots 2 and 3 containing spectra for all 6 positions tested).

2. Bearing Health - No bearing frequencies which would indicate defects were found. Therefore, this indicated motor bearings were likely in good condition.

3. Balance and Alignment - All spectra indicated a well balanced rotor assembly with good alignment.

4. Electrical Condition - No electrical problems were evident from the spectra based on signatures taken in the morning with a lighter gauge metal product, nor were any evident in the afternoon when a heavier gauge, non-annealed product was run which put the rolling mill drive under heavier strain.

5. Mechanical Looseness - Definite mechanical looseness was indicated in spectra taken at all 6 motor positions, based on the presence of a

© Copyright 2001 Technical Associates of Charlotte, P.C.

series of motor speed harmonics (multiples of RPM). On the March 15 test day, it was verbally reported that the source of looseness was likely either between the inboard bearing surface and shaft journal or at the female bearing seat supporting the housing. Since highest overall levels on March 15 were all below .20 in/sec (compared to .30 in/sec Alarm level), there was no indication of imminent failure at that time. However, because plant maintenance engineering reported much higher amplitudes had been present on March 12, and due to the presence of pronounced pulsing noises each time the motor was brought up to speed, concern was expressed on the health of the machine despite the fact that vibration was now, on March 15, below alarm.

B. MILL 4 REWIND MOTOR (See Figure 2)

1. In general, vibration was about 5 times higher on March 15, 1988 than it was when similar spectra were captured on the #4 Inboard Rewind Motor on December 10, 1987. All spectra were dominated by the presence of bearing sum and difference frequencies indicating defects on the outer race. Vibration spectra indicated motor bearings in advanced state of wear. On March 15, it was recommended that both motor bearings be replaced ASAP.

2. The motor still seemed well balanced. There was no evidence of electrical problems at that time. However, slight misalignment was indicated between the outboard and inboard motors and/or there seemed to be a slight problem with the electric clutch assembly located between the two motors (see Figure 2).

III. RECOMMENDATIONS (Made before failure of the Mill 4 Main Drive Motor)

A. MILL 4 MAIN DRIVE MOTOR

Because of the definite mechanical looseness symptoms described in “Conclusions” Section II-A5, it was recommended that the plant take readings each day to monitor possible deterioration of the problem, and also to see if any defect frequencies for the motor bearings would appear. It was not felt that any significant damage had yet been done to the bearings due to the complete absence of defect frequencies (this could be asserted with particular confidence since defect frequencies for this particular SKF 6236 bearing were well removed from integer running speed harmonics - BPFI = 6.338X; BPFO = 4.662X; BSF = 3.205X). In fact, a series of zoom spectra confirmed that no noticeable frequencies other than running speed harmonics and gear mesh frequencies were present. It was further recommended that measurement intervals could be increased to weekly and later monthly if no further damage was evident in future vibration spectra.

UPDATE:

Later, it was reported that the Mill 4 Main Drive Motor did experience failure on March 18. Inspection confirmed the mechanical looseness, and found the source at the inboard motor bearing, revealing that the shaft journal had been turning in the bearing removing .126" from the shaft diameter. In all probability, much of

© Copyright 2001 Technical Associates of Charlotte, P.C.

this damage was done on March 12 when all the high vibration and noise was noticed by the plant. The inspection also confirmed that the bearing itself was still in fact in good shape as had been indicated by the vibration signatures, despite the excessive damage to the shaft.

B. MILL 4 REWIND MOTOR

It was strongly recommended that the plant take immediate steps to replace the motor bearings due to both the amplitude and the presence of a great number of the difference frequencies spaced at the SKF 6218 outer race defect frequency (4.11X RPM). In addition, it was recommended that the plant carefully check the alignment of the rewind drive assembly as well as the condition of the electric clutch assembly located between the outboard and inboard motors (see Figure 2).

IV. DISCUSSION

A. MILL 4 MAIN DRIVE MOTOR

A series of axial, horizontal and vertical measurements were made on the main drive motor bearings in order to assess the condition of the motor (see Figure 1). The vibration spectra derived from these measurements indicated the following:

1. No vibration levels exceeding alarm were present on either motor bearing on this test day.

2. No bearing defect frequencies were present, indicating good bearing condition.

3. Balance and alignment indications were good.

4. No motor electrical problems were present, even when the mill was operating with heavy gage material.

5. The dominant feature of all the spectra taken was the large number of harmonics of running speed, an indication of mechanical looseness in the motor. A verbal diagnosis at the plant was made that the most likely problem was looseness of the inboard bearing seat in which the bearing housing was held (closest to gearbox) and/or looseness between the bearing surface and shaft journal.

A visual inspection of the inboard bearing was made with no concrete conclusions, but there was some indication that the bearing seat could be loose.

Later, it is understood that a shutdown was mandated on March 18 because of sparks and other signs of imminent failure. Disassembly and inspection by the plant revealed that on the inboard end of the motor, the shaft journal had been turning in the bearing and the journal had been eroded to the point of being .126 inch undersized. The journal, normally a press fit on the bearing, now had a very large clearance, leaving the motor shaft loose on the inboard end. This explains the mechanical looseness indicated by the March 15 measurements, an example of which is shown by Plot 1, derived from a measurement at the inboard

© Copyright 2001 Technical Associates of Charlotte, P.C.

bearing of the motor. Note the large number of harmonics of running speed present in the spectrum, a typical spectrum for mechanical looseness. For comparison purposes, Plots 2 and 3 are included showing spectra taken at all 6 motor positions located on the outboard and inboard bearing housings. Note that mechanical looseness is evident in all 6 spectra, based on the large number of 567 RPM running speed harmonics. However, note that all overall levels remained below the alarm level for this machine, which would be about **.300 in/sec** (overalls ranged from .074 in/sec to .176 in/sec).

The motor was disassembled and the shaft was rebuilt by General Electric Service Shop. It is understood that the motor has been put back into service and is apparently operating satisfactorily. It is recommended that signatures be obtained to check current machine health and to serve as **Baseline Signatures** for future reference.

B. MILL 4 REWIND MOTOR

Measurements made on the Mill 4 Inboard Rewind Motor indicated definite bearing problems with the SKF 6218 outboard bearing. While measurements were being made, a large number of difference frequencies were noticed. The spacing of these difference frequencies was equal to the outer race defect frequency (4.11 F). The upper spectrum of Plot 4 shows a high overall of .544 in/sec, dominated by the outer race difference frequencies. The lower signature is a zoom spectrum showing that the difference frequencies were spaced at 2280 CPM when the motor speed was 555 RPM - equal to 4.11 times the motor speed confirming the presence of bearing wear. For comparison purposes, spectra taken at all 6 positions are provided in Plots 5 and 6. They likewise showed wear of the outboard bearing due to high outer race frequency difference frequencies.

Therefore, on the March 15 test day, it was recommended that both of the bearings be changed in the inboard rewind motor, a recommendation which has been carried out. The motor is now back in service and is apparently functioning well.

V. CORRECTIVE ACTIONS TAKEN

A. MILL 4 MAIN DRIVE MOTOR

Upon investigation, the plant found that one of the problems which may have contributed to the bearing failure was too light a press fit. This bearng had a bore of 180 mm (7.086 in), but a press fit of only .0009 inch. It was decided to increase this to an M6 fit which ranges from .0006 to .0030 inch in the new motor bearings. In addition, an investigation also found that when this variable speed machine ran at certain speeds, the gear mesh frequency transmitting from the gearbox into the motor was exciting an axial natural frequency of the motor. Thus, it was decided not to operate the motor in this particular speed range in which it was resonant.

© Copyright 2001 Technical Associates of Charlotte, P.C.

B. MILL 4 REWIND MOTOR

Each of the motor bearings was first replaced. Then, a slight misalignment problem was corrected.

VI. RESULTS OF CORRECTIVE ACTIONS

Vibration spectra were captured with an IRD 890 analyzer after corrective actions were completed as part of a plant predictive maintenance program.

A. MILL 4 MAIN DRIVE MOTOR

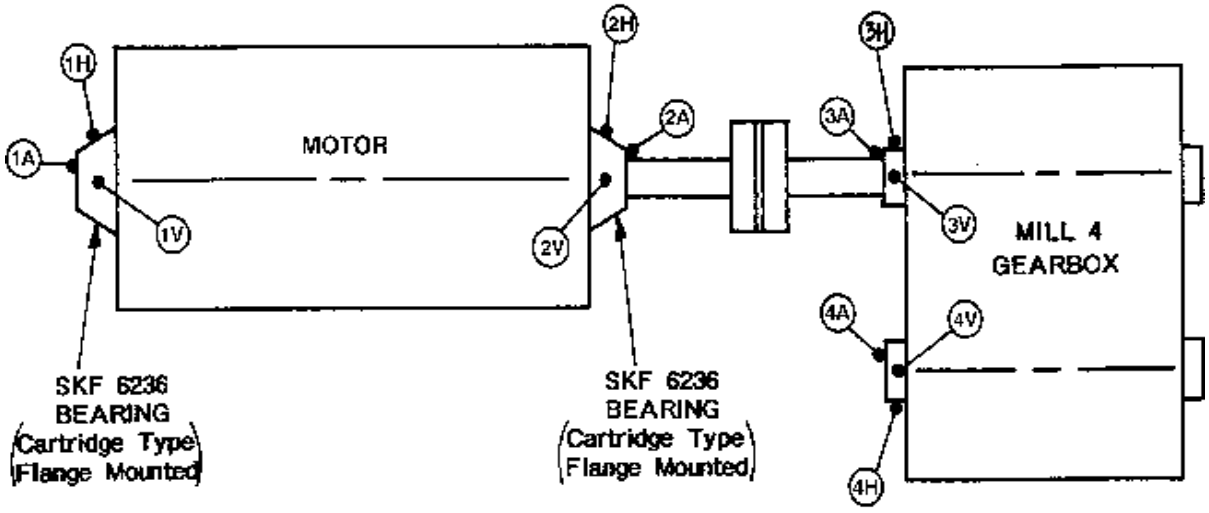
Plots 7A and 7B compare spectra taken before and after motor bearing replacement respectively (note that the PMP route spectra had an F_{MAX} of 60,000 CPM compared to 12,000 CPM on the Plot 7A spectrum taken during the earlier diagnostic investigation). Importantly, note that no harmonics at running speed were present in the "After" spectrum of Plot 7B (at least none were present exceeding the .004 in/sec threshold value). Note also that the overall velocity dropped from .176 in/sec to .059 in/sec.

B. MILL 4 REWIND MOTOR

Plots 8A and 8B compare "Before" and "After" spectra on the Mill 4 Rewind Motor. Note the dramatic drop in overall velocity from .544 in/sec to only .018 in/sec - a reduction of over 96%. Also, note the tremendous drop in higher frequency vibration, particularly between 20,000 and 60,000 CPM.

© Copyright 2001 Technical Associates of Charlotte, P.C.

FIGURE 1. MILL 4 MAIN DRIVE ARRANGEMENT



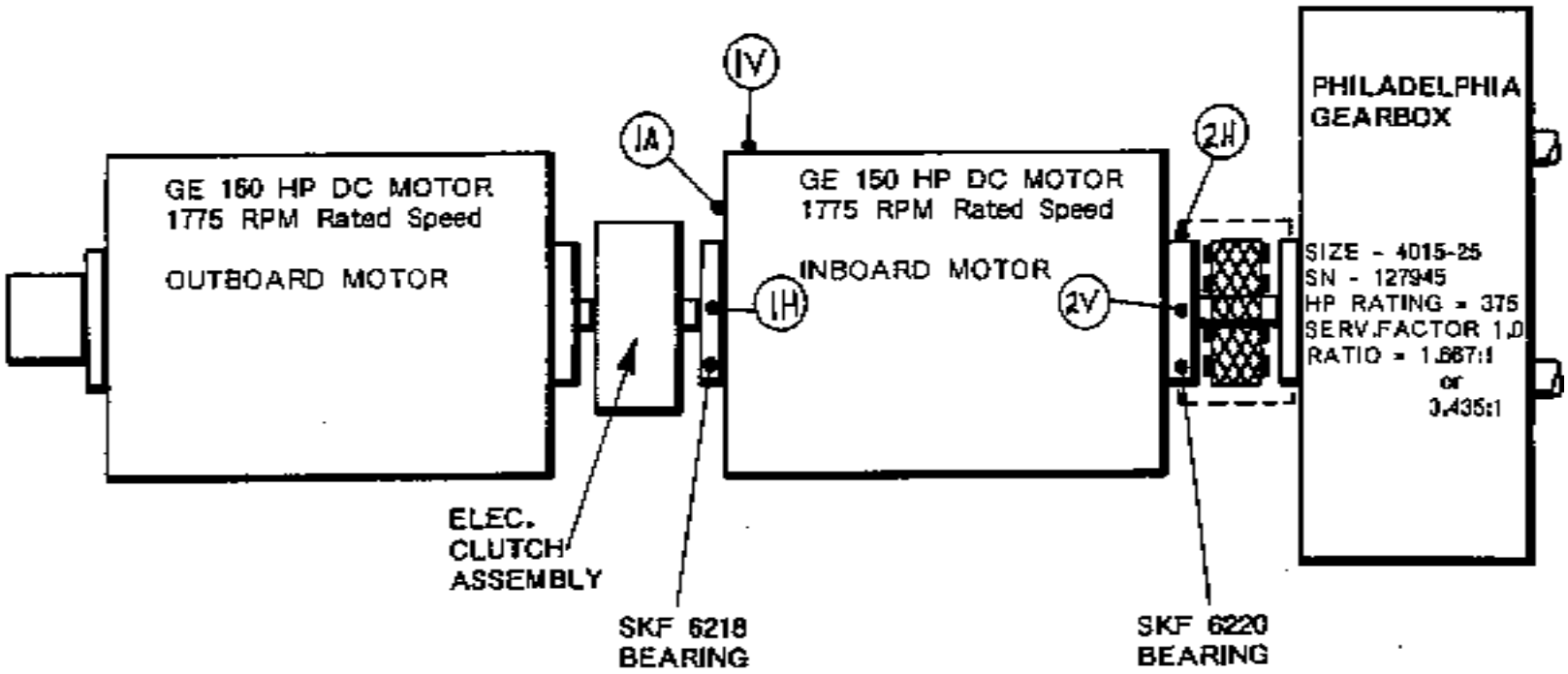
SKF 6236 BEARING DEFECT FREQUENCIES AT 567 RPM:

BPFI = 6.338 X RPM = 6.338 X 567 = 3594 CPM
BPFO = 4.662 X RPM = 4.662 X 567 = 2643 CPM
BSF = 3.205 X RPM = 3.205 X 567 = 1817 CPM
FTF = 0.424 X RPM = 0.424 X 567 = 240 CPM

© Copyright 2001 Technical Associates of Charlotte, P.C.

© Copyright 2001 Technical Associates of Charlotte, P.C.

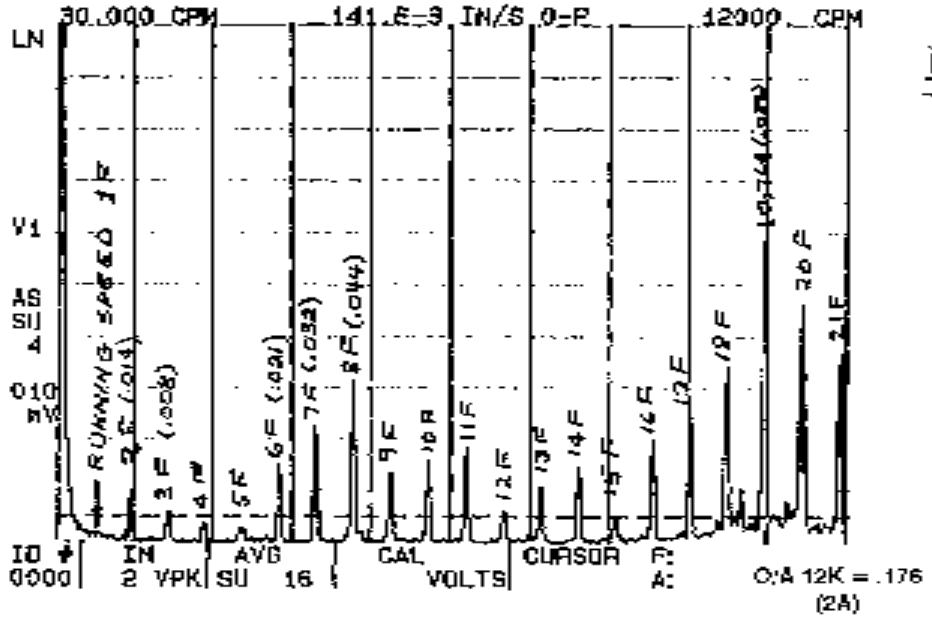
FIGURE 2. MILL 4 REWIND DRIVE ASSEMBLY



MILL 4 REWIND MOTOR BEARING DEFECT FREQUENCIES AT 555 RPM

BEARING	BPFI	BPFO	BSF	FTF
SKF 6220	5,907 X = 3278 CPM	4.093 X = 2272 CPM	2.665 X = 1479 CPM	0.409 X = 227 CPM
SKF 6218	5,889 X = 3268 CPM	4.111 X = 2282 CPM	2.723 X = 1511 CPM	0.411 X = 228 CPM

© Copyright 2001 Technical Associates of Charlotte, P.C.



PEAK PICK ON TRACE 1

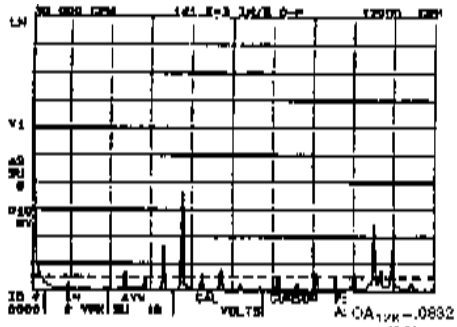
RPM = 567

POINT 2A ON MOTOR
ALL SIGNIFICANT SPIKES
ARE HARMONICS OF
RUNNING SPEED
MECHANICAL LOOSNESS
IS INDICATED

IN/S 0-P	CPM	
13.5E-3	1140.0	= 2F
8.37E-3	1710.0	= 3F
21.2E-3	3390.0	= 6F
32.2E-3	3959.9	= 7F
44.4E-3	4530.0	= 8F
19.3E-3	5100.0	= 9F
22.5E-3	5670.0	= 10F
25.8E-3	5240.0	= 11F
8.37E-3	6810.0	= 12F
15.5E-3	7350.0	= 13F
20.6E-3	7920.0	= 14F
7.08E-3	8490.0	= 15F
28.3E-3	9060.0	= 16F
34.8E-3	9630.0	= 17F
48.3E-3	10200.	= 18F
7.08E-3	10320.	
14.8E-3	10410.	
83.0E-3	10769.	= 19F
7.73E-3	10919.	
11.6E-3	11100.	
9.01E-3	11159.	

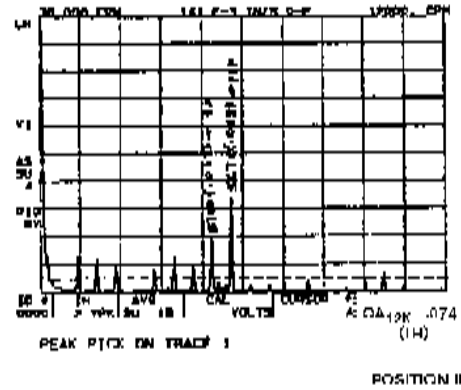
PLOT 1

MILL 4 MAIN DRIVE MOTOR



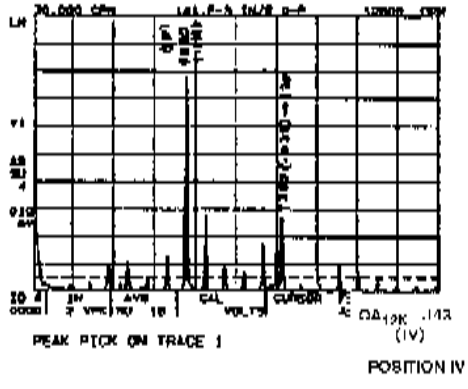
IN/S 0-P	CPM
14.2E-3	210.00
7.08E-3	330.00
10.3E-3	2820.0
23.8E-3	3959.9
51.5E-3	4530.0
8.37E-3	5100.0
10.9E-3	5670.0
9.66E-3	8490.0
9.01E-3	9630.0
35.4E-3	10200.
11.6E-3	10410.
21.2E-3	10769.

MIL4 MDMOT



IN/S 0-P	CPM
19.3E-3	240.00
7.08E-3	360.00
18.7E-3	1140.0
17.4E-3	1710.0
13.5E-3	2250.0
11.6E-3	3390.0
18.0E-3	3959.9
14.2E-3	4530.0
27.0E-3	5100.0
47.6E-3	5670.0
10.3E-3	10200.

MIL4 MDMOT

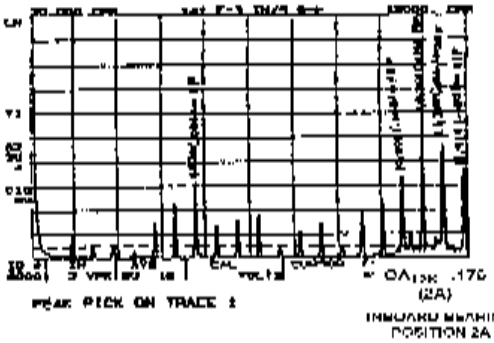


IN/S 0-P	CPM
13.5E-3	2250.0
15.5E-3	2820.0
7.73E-3	3390.0
18.0E-3	3959.9
111.E-3	4530.0
39.3E-3	5100.0
13.5E-3	5670.0
10.3E-3	6240.0
24.5E-3	6810.0
34.1E-3	7200.0
38.6E-3	7379.7
13.5E-3	9060.0
9.01E-3	9630.0
8.37E-3	10200.

MILL 4 MAIN DRIVE MOTOR
POSITIONS IA,IH,&IV
PLOT 2

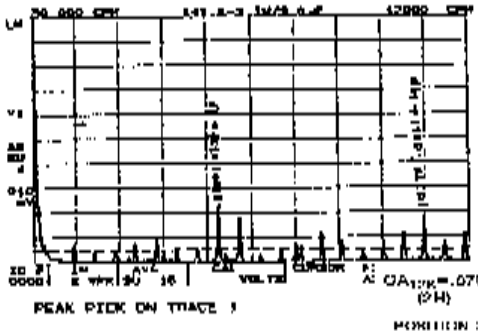
MIL4 MDMOT

© Copyright 2001 Technical Associates of Charlotte, P.C.



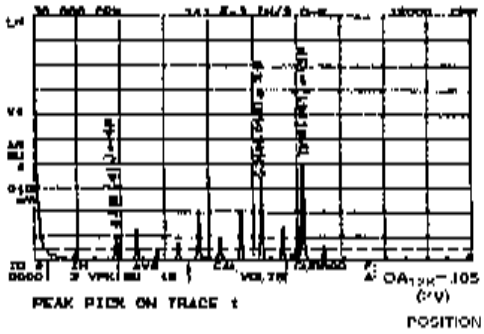
IN/S 0-P	CPM	
13.5E-3	1140.0	=2F
8.37E-3	1710.0	-3F
21.2E-3	3380.0	-6F
32.2E-3	3959.9	=7F
44.4E-3	4530.0	=8F
19.3E-3	5100.0	9F
22.5E-3	5670.0	-10F
25.8E-3	6240.0	11F
8.37E-3	6810.0	12F
15.5E-3	7350.0	13F
20.6E-3	7920.0	-14F
7.08E-3	8490.0	-15F
28.3E-3	9060.0	=16F
34.8E-3	9630.0	17F
48.3E-3	10200.	18F
7.08E-3	10320.	
14.8E-3	10410.	
83.0E-3	10769.	=19F
7.73E-3	10919.	
11.6E-3	11100.	
9.01E-3	11159.	

MILL 4 MAIN DRIVE MOTOR



IN/S 0-P	CPM	
12.2E-3	240.00	
10.3E-3	300.00	
7.08E-3	360.00	
10.9E-3	1140.0	=2F
10.9E-3	2820.0	4F
12.9E-3	3390.0	5F
7.08E-3	3600.0	
7.73E-3	3959.9	7F
7.08E-3	4530.0	=8F
31.5E-3	5100.0	=9F
25.1E-3	5670.0	=10F
7.73E-3	6810.0	=12F
10.9E-3	7200.0	
9.01E-3	7350.0	=13F
17.4E-3	7920.0	-14F
17.2E-3	8490.0	=15F
7.08E-3	9060.0	=16F
10.9E-3	9630.0	-17F
16.7E-3	10200.	-18F
26.4E-3	10769.	19F
10.9E-3	11340.	20F

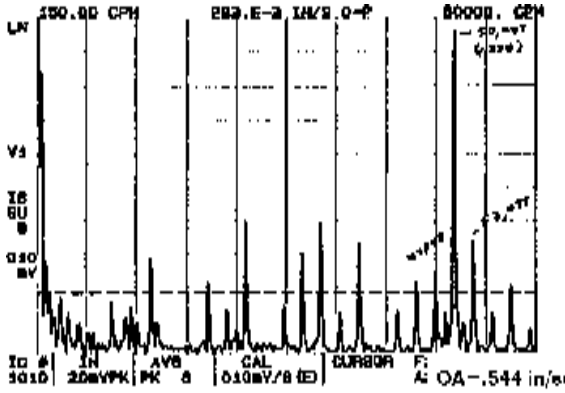
MILL 4 MAIN DRIVE MOTOR



IN/S 0-P	CPM	
14.2E-3	240.00	
15.5E-3	2250.0	4F
18.7E-3	2820.0	5F
10.3E-3	3959.9	-7F
28.3E-3	4530.0	8F
12.2E-3	5100.0	9F
27.7E-3	5670.0	=10F
47.0E-3	6240.0	11F
20.0E-3	6810.0	12F
20.8E-3	7200.0	
57.3E-3	7379.7	13F
9.66E-3	7920.0	=14F

MILL 4 MAIN DRIVE MOTOR
POSITIONS 2A,2H,&2V
PLOT 3

© Copyright 2001 Technical Associates of Charlotte, P.C.

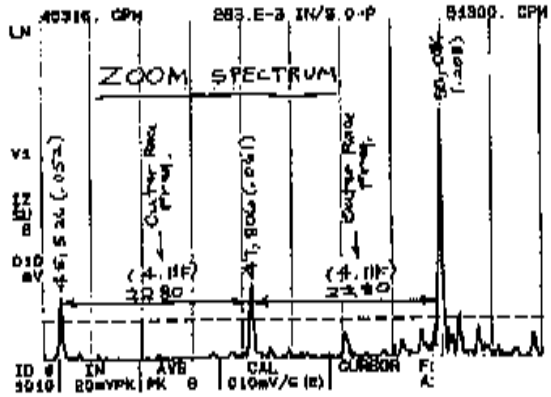


PEAK PICK ON TRACE 1

IN/S 0-P	CPM
239.E-3	600.00
80.1E-3	1200.0
53.0E-3	1649.9
82.7E-3	13649.
60.7E-3	20550.
114.E-3	25049.
86.6E-3	31950.
112.E-3	34200.
94.3E-3	38850.
60.7E-3	45600.
68.5E-3	47848. Δ=2248
274.E-3	50248. Δ=2400
96.9E-3	52498. Δ=2250
56.9E-3	57000.

POSITION IA
OUTBOARD BEARING
(INBOARD MOTOR)
MOTOR SPEED = 555 RPM
(AT THIS INSTANT)

MIL4 REWMT



PEAK PICK ON TRACE 1

IN/S 0-P	CPM
51.7E-3	45526.
60.7E-3	47806.
208.E-3	50086.
34.9E-3	50310.

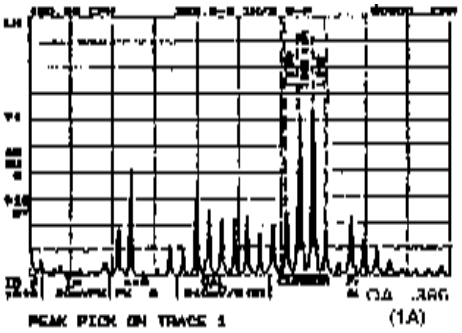
POSITION IA
CURRENT MOTOR SPEED = 555 RPM (AT THIS INSTANT)

$2280 CPM - \frac{2280}{555} - 4.11F$ – OUTER RACE BEARING FREQUENCY

MILL 4 REWIND MOTOR
POSITION IA
PLOT 4

MIL4 REWMT

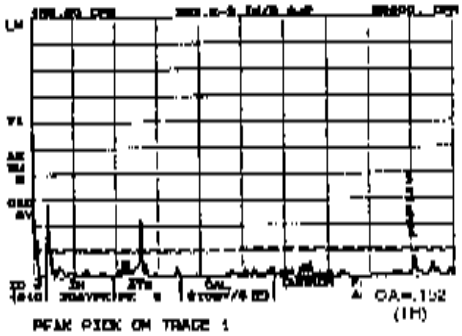
© Copyright 2001 Technical Associates of Charlotte, P.C.



POSITION IA
MOTOR SPEED = 460 RPM
(TOWARDS END OF CYCLE)
(SIDEBANDS = 1800 CPM)
MILL 4 REWIND MOTOR

IN/S 0-P	CPM	
51.7E-3	12899.	
115.E-3	14700.	
81.4E-3	23849.	
71.1E-3	25649.	
60.7E-3	27600.	
62.0E-3	29399.	
64.6E-3	31199.	
46.5E-3	33000.	
56.9E-3	34950.	
69.8E-3	36750.	Δ=1800
175.E-3	38850.	
36.2E-3	39000.	Δ=1650
182.E-3	40350.	
40.1E-3	40800.	Δ=1350
50.4E-3	42150.	Δ 3750
64.6E-3	45898.	Δ=1802
38.8E-3	47700.	

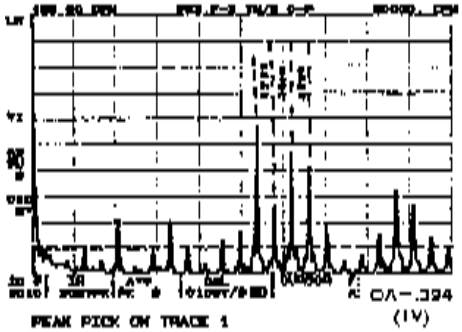
MIL4 REWMT



POSITION IH

IN/S 0-P	CPM
42.6E-3	600.00
81.4E-3	2250.0
63.3E-3	15449.

MIL4 REWMT



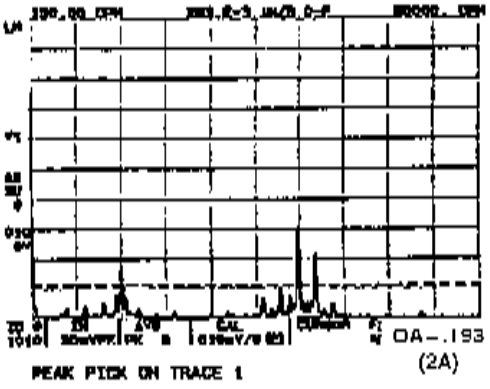
POSITION IV

IN/S 0-P	CPM	
96.9E-3	600.00	
32.3E-3	1200.0	
60.7E-3	12450.	
58.1E-3	19800.	
37.5E-3	27300.	
45.2E-3	29699.	
31.0E-3	31650.	
164.E-3	32249.	
74.9E-3	34800.	Δ=2550
133.E-3	37200.	Δ=2400
118.E-3	39750.	Δ 2550
53.0E-3	42150.	
43.9E-3	49650.	
93.0E-3	52050.	
74.9E-3	54598.	
98.8E-3	57000.	
26.2E-3	7500	
23.8E-3	17400	

MIL4 REWMT

MILL 4 REWIND MOTOR
(POS. IA,IH,&IV)
PLOT 5

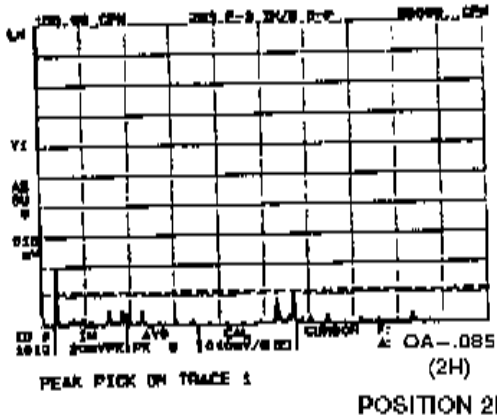
© Copyright 2001 Technical Associates of Charlotte, P.C.



IN/S 0-P	CPM
45.2E-3	11850.
86.6E-3	35700.
60.7E-3	38100.
19.9E 3	30900
26.5E-3	33300

POSITION 2A

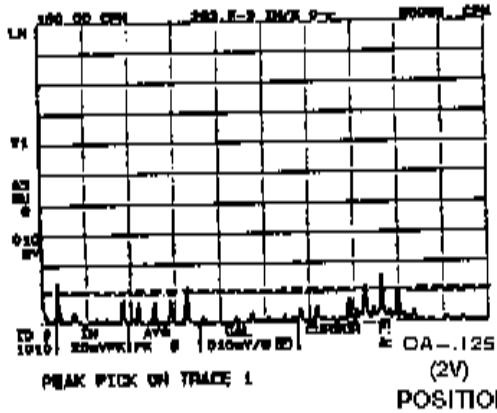
MIL4 REWMT



IN/S 0-P	CPM
55.6E-3	2250.0
26.4E-3	31800
27.9E-3	34050

POSITION 2H

MIL4 REWMT



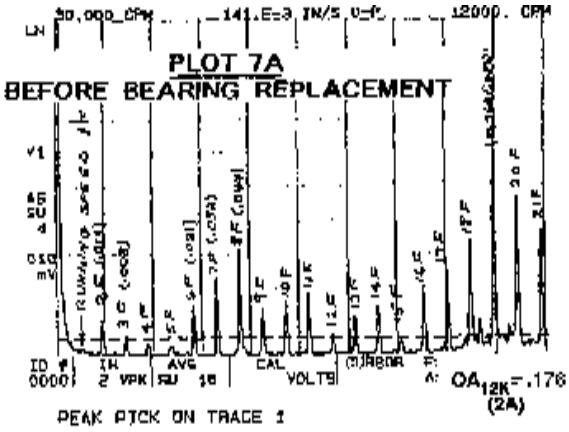
IN/S 0-P	CPM
38.8E-3	2250.0
33.6E-3	19650.
33.6E-3	43498.
43.9E-3	45750.
31.0E-3	47848.
22.4E-3	10950
19.4E 3	13050
19.5E 3	15300
22.4E 3	17400
34.1E 3	19650
23.2E 3	41400

POSITION 2V

MILL 4 REWIND MOTOR
(POS. 2A,2H,&2V)
PLOT 6

MIL4 REWMT

© Copyright 2001 Technical Associates of Charlotte, P.C.



PLOT 7A
BEFORE BEARING REPLACEMENT

IN/S 0-P	CPM
13.5E-3	1140.0
8.37E-3	1710.0
21.2E-3	3390.0
32.2E-3	3959.9
44.4E-3	4530.0
19.3E-3	5100.0
22.5E-3	5670.0
25.8E-3	6240.0
8.37E-3	6810.0
15.5E-3	7350.0
20.6E-3	7920.0
7.08E-3	8490.0
28.3E-3	9060.0
34.8E-3	9630.0
48.3E-3	10200.
7.08E-3	10320.
14.8E-3	10410.
83.0E-3	10769.
7.73E-3	10919.
11.6E-3	11100.
9.01E-3	11159.

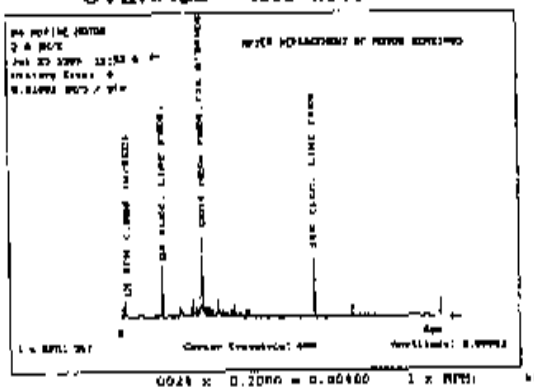
MILL 4 MAIN DRIVE MOTOR

(REPEAT OF PLOT 1)
RPM = 357

POINT 2A ON MOTOR
ALL SIGNIFICANT SPIKES
ARE HARMONICS OF
RUNNING SPEED

MECHANICAL LOOSENESS
IS INDICATED

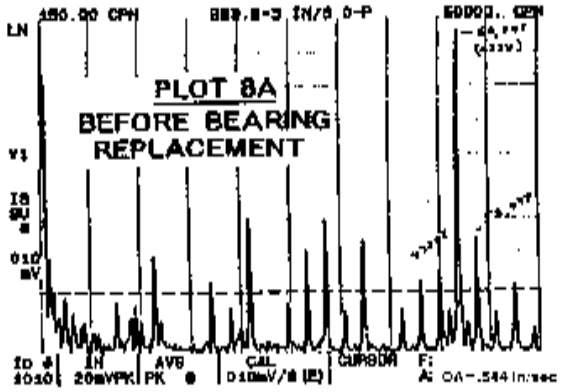
PLOT 7B
AFTER REPLACEMENT OF MOTOR BEARINGS
OVERALL = .059 in/sec



Frequency	1 x RPM	Amplitude	Frequency	x RPM	Amplitude
150	0.56	0.00550	36000	91.49	0.00650
600	1.06	0.00601	43200	76.19	0.00648
7200	12.70	0.01701	59850	106.66	0.01048
10800	18.52	0.00460			
10950	19.31	0.00480			
12900	22.75	0.00550			
13600	23.81	0.00649			
14700	25.73	0.00549			
15300	26.98	0.00550			
15900	28.04	0.00648			
16500	29.10	0.00448			
17940	30.95	0.00548			
18450	32.54	0.00448			
19560	35.19	0.00448			
20550	36.34	0.00750			
22800	40.21	0.00499			
23400	41.27	0.00599			

T - Threshold adjust Q - quit

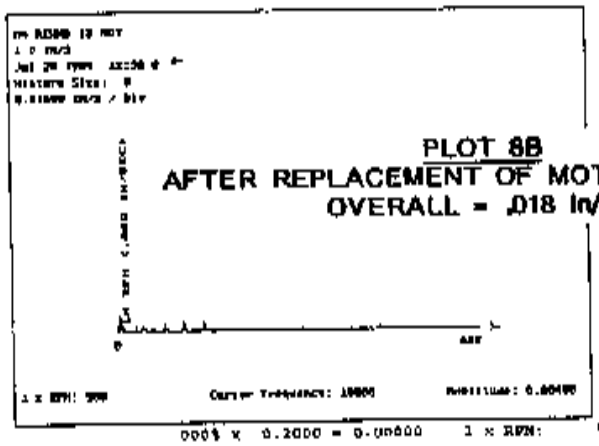
© Copyright 2001 Technical Associates of Charlotte, P.C.



IN/S 0-P	CPM	
239.E-3	600.00	
80.1E-3	1200.0	
53.0E-3	1649.9	
82.7E-3	13649.	
60.7E-3	20550.	
114.E-3	25049.	
86.6E-3	31950.	
112.E-3	34200.	
94.3E-3	38850.	
60.7E-3	45600.	
68.5E-3	47848.	△ -2248
274.E-3	50248.	△ -2400
96.9E-3	52498.	△ =2250
56.9E-3	57000.	

(REPEAT OF PLOT 4)

POSITION IA
OUTBOARD BEARING
(INBOARD MOTOR)
MOTOR SPEED = 555 RPM
(AT THIS INSTANT)



Machine: HS REWND IE NOT | Pos: 1 | Dir: V | Units: IN/S | Date: Jul 25 1999 12:50

Frequency	1 X RPM	Amplitude	Frequency	1 X RPM	Amplitude
150	0.30	0.00599	43200	86.40	0.00248
600	1.20	0.00200			
900	1.80	0.00055			
3000	6.00	0.00200			
4050	8.10	0.00249			
5550	11.10	0.00098			
5850	11.70	0.00058			
7200	14.40	0.00149			
7650	15.30	0.00349			
10800	21.60	0.00400			
13950	27.90	0.00400			
14400	28.80	0.00098			
19200	38.40	0.00098			
21600	43.20	0.00098			
22050	44.10	0.00098			
30000	60.00	0.00098			
30450	60.90	0.00149			

T - Threshold adjust 0 - Quit

© Copyright 2001 Technical Associates of Charlotte, P.C.

DETECTION OF A BALL BEARING PROBLEM ON A GLAND STEAM EXHAUST FAN WHOSE BEARING MANUFACTURER & MODEL NUMBER WERE UNKNOWN

I. BACKGROUND

A utility company contracted Technical Associates to provide them a complete vibration analysis predictive maintenance program. This program began by acquisition of baseline measurements on all machines selected followed by data analysis and composition of a **Baseline Report**. For the most part, little was known about the machines except for nameplate data. One of the machines tested was the Gland Steam Exhaust Fan. This unit was an integrally mounted fan (shaft mounted) with anti-friction bearings and a nominal running speed of 3500 RPM. The machine arrangement is shown in Figure 1.

II. DISCUSSION OF RESULTS

IRD Mechanalysis Data Analysis System
Dec-04-89

TABLE I.

Machine Report (Latest Data) for: GLND STM EXH 61

Date	Time	Pos	Dir	Ampl	Units	Change	Alarm	Inspection
Aug-24-89	15:00	1	A	.286	IN/S	-----%	.375	
Aug-24-89	15:00	1	H	.171	IN/S	-----%	.375	
Aug-24-89	15:00	1	H	1.14	g/SE	-----%	1.25	
Aug-24-89	15:00	1	O	.176	IN/S	-----%	.375	
Aug-24-89	15:00	1	V	.211	IN/S	-----%	.375	
Aug-24-89	15:02	2	H	.208	IN/S	-----%	.375	
Aug-24-89	15:02	2	H	*1.32	g/SE	-----%	1.25	
Aug-24-89	15:02	2	V	*.401	IN/S	-----%	.375	

8 lines this report

Table I shows the list of overall levels of vibration (in/sec) and spike energy (g/SE). Note from Table I that both spike energy (g/SE) measurements on the outboard and inboard bearings were relatively high (1.14 g/SE on the outboard Position 1 bearing and 1.32 g/SE on the inboard bearing Position 2). The Position 2H spike energy exceeded the alarm as did the Position 2 vertical (2V) overall velocity measurement (.401 in/sec compared to an alarm of .375 in/sec). The Position 1V spectrum (Figure 2) had a well defined frequency at 76,200 CPM with running speed (3600 CPM nominal) sidebands. The Position 2V spectrum shown in Figure 3 had a moderately large 1X RPM amplitude of .350 in/sec and the same 76,200 CPM frequency, again with running speed sidebands.

© Copyright 2001 Technical Associates of Charlotte, P.C.

III. CONCLUSIONS

A. The 76,200 CPM frequency seen on both motor bearings was likely the natural frequency of one of the bearing components (such bearing component natural frequencies are commonly found between approximately 30,000 and 120,000 CPM). When such bearing natural frequencies are sufficiently excited to cause them to appear in the velocity spectrum, this normally indicates the bearing is passing through the second stage of wear (such rolling element bearings usually have four rather well defined wear stages until eventual failure). Normally, when 1X RPM sidebands then appear around the bearing natural frequency, this signals a more serious problem and that the bearing is nearing the end of Stage 2. At this point, the bearing usually has only about 5% remaining life.

B. The fan wheel is unbalanced.

IV. RECOMMENDATIONS

A. Since the signatures on both bearings indicated late Stage 2 wear, and because spike energy levels were likewise high on each, it was recommended that both motor bearings be replaced.

B. It was then recommended the plant re-evaluate the unbalance condition after the bearings were replaced. If both signature and phase analysis then still indicated unbalance, the plant was instructed to clean and balance the fan wheel (see Figure 4).

© Copyright 2001 Technical Associates of Charlotte, P.C.

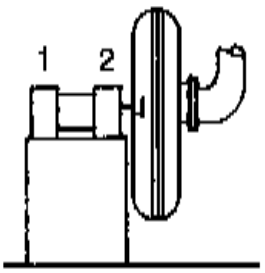
Technical Associates of Charlotte, P.C.
(704-333-8011)

MACHINERY DATA GUIDE SHEET

RELIABILITY SURVEY — predictive maintenance program — (Page__of__)

UNITS
51
52
(61)
62

DETECTION OF A BALL BEARING PROBLEM ON A GLAND STEAM EXHAUST FAN WHOSE BEARING MANUFACTURER AND MODEL NUMBER WERE UNKNOWN



MACHINE IDENT: GLAND STEAM EXHAUST FAN

AREA: _____

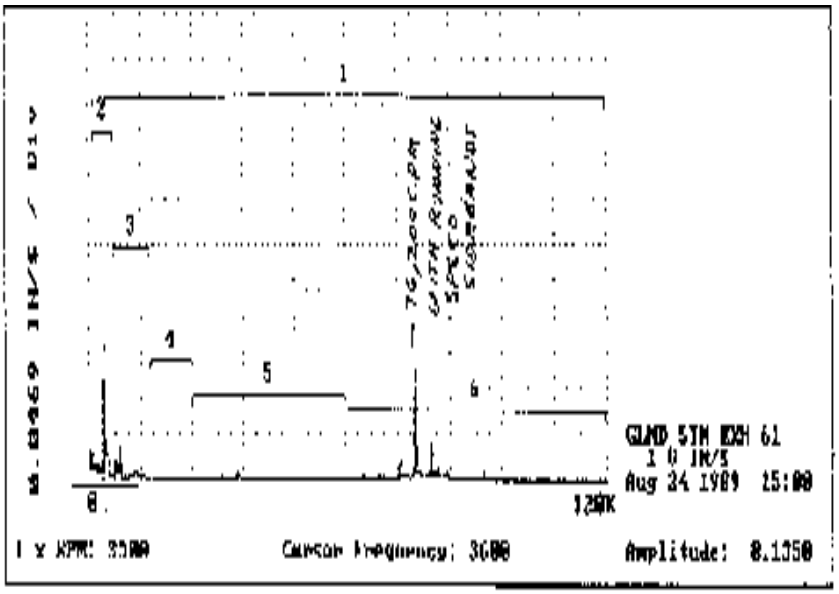
ROUTE: _____

DRIVER	INTERMEDIATE	DRIVEN
RPM min : 3500	RPM In :	RPM min : 3500
max :	out :	max :
MFGR :	RATIO:	MFGR. :
MODEL :	MODEL :	MODEL :
SER NO. :	SER NO. :	SER NO. :
V____ P____ A____	RATING :	RATING :
SHEAVE : ____ DIA.	BELT # :	#BLADES :
BRG#: A (I)	BRG#: (I)	BRG#: A (I)
BRG#: (O)	BRG# (O)	BRG#: (O)
H.P.: 10	GEAR T: ____ CT.	GEAR T: ____ CT.
OTHER:	OTHER:	OTHER:

FIGURE 1

IF-08

© Copyright 2001 Technical Associates of Charlotte, P.C.



602% x 0.500 = 0.0100 1 x RPM: 3500.00

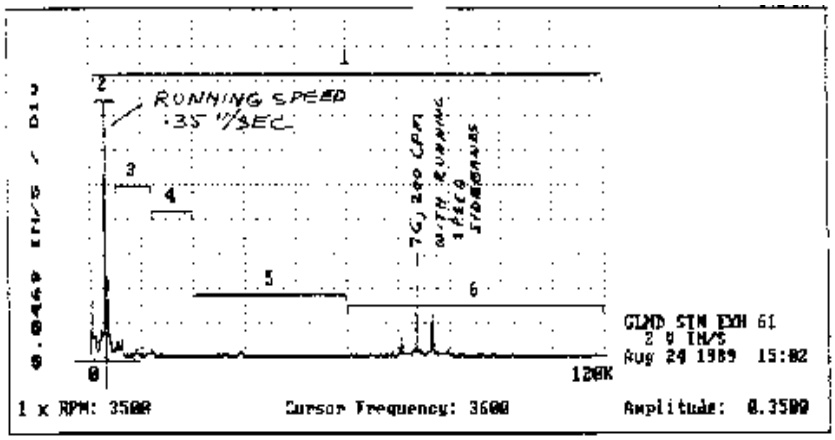
Machine	Pos	Dir	Units	Date
GLND STM EXH 61	1	[illegible]	IN/S	Aug 24 1989 15:00

Frequency	1 x RPM	Amplitude
400	0.09	0.0300
1200	0.34	0.0250
2100	0.60	0.0150
3600	1.03	0.1350
6000	1.71	0.0175
7200	2.06	0.0375
10800	3.09	0.0100
72600	20.74	0.0300
78200	21.77	0.1100
79500	22.71	0.0400
81300	23.23	0.0125
82400	23.53	0.0100

T = Threshold Adjust Q = Quit

FIGURE 2

© Copyright 2001 Technical Associates of Charlotte, P.C.



002% x 0.500 = 0.0100 1 x RPM: 3500.00

Machine	Pos	Dir	Units	Date
GLND STM EXH 61	2	V	IN/S	Aug 24 1989 15:02

Frequency	1 x RPM	Amplitude
700	0.09	0.0775
2400	0.69	0.0350
3600	1.03	0.3500
6300	1.80	0.0200
7200	2.06	0.0225
14400	4.11	0.0100
72600	20.74	0.0250 - 1X RPM S.B.
76200	21.77	0.1050 ←
79800	22.80	0.0575 — 1X
83400	23.83	0.0100 — 2X

A-3600
A-3600

T - Threshold adjust Q - Quit

FIGURE 3

© Copyright 2001 Technical Associates of Charlotte, P.C.

FIGURE 4.

R-9210-1

© Copyright 2001 Technical Associates of Charlotte, P.C.

SIGNIFICANT IMPROVEMENT IN MOTOR AND BLOWER VIBRATION DUE TO BELT REPLACEMENT

I. BACKGROUND

During a Predictive Maintenance Program vibration survey by Technical Associates on March 22, 1989, a belt-driven air handler, 304 AIRHAND 004, was detected as having an overall vibration of 1.34 in/sec on Position 2H. Position 2H is the inboard motor bearing horizontal measurement as seen in the machine sketch of Figure 1. Note the Figure 1 spectrum for Position 2H on March 22, 1989 shows the dominant vibration was .930 in/sec at 450 CPM. Motor speed was a nominal 1800 RPM and fan speed a nominal 1050 RPM as verified by a hand-held tachometer measurement. It was determined that 2X belt frequency approximated 450 CPM and was the likely cause of the high amplitude. A report of this finding was issued on 3/22/89 warning of the high vibration and its possible causes as seen in Figure 2 which is a copy of the **Rank-Ordered Results and Recommendations Report** issued to the client by Technical Associates. Unfortunately, the client did not follow the recommendations given in Figure 2, but simply aligned and tensioned the belts as seen by the 5/23/89 spectrum taken during the next survey. Of course, the 450 CPM belt frequency second harmonic was still present dominating the spectrum. Again a report was issued recommending replacement of the belt followed by proper tensioning and alignment of the motor and fan sheeves.

II. CONCLUSIONS

A. The 450 CPM amplitude was the result of a belt and/or sheave problem.

III. RECOMMENDATIONS

A. Replace the belts which are the dominant vibration source in this machine.

B. Check the fan sheave for excessive runout and repair/replace as needed.

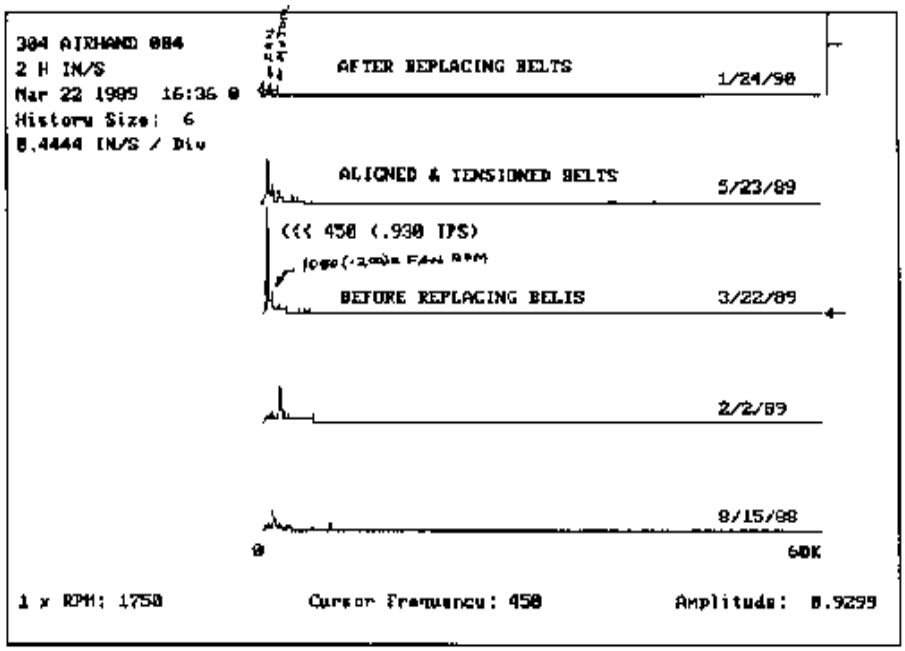
C. Align and tension the belts.

IV. RESULTS OF CORRECTIVE ACTIONS

As seen in the waterfall spectra of Figure 1 and the Position 2H Trend Graph of Figure 3, the alignment and tensioning of the belts reduced vibration significantly, but the 450 CPM vibration remained. Later, the belts were replaced resulting in the spectrum dated January 24, 1990, showing no 450 CPM vibration as noted in the frequency/order/amplitude tabular listing near the middle of Figure 1 that is also dated January 24, 1990. The Trend Graph of Figure 3 shows the improved results which occurred to the Position 2H overall vibration as a result of these corrective actions.

CF7 R-9210-1

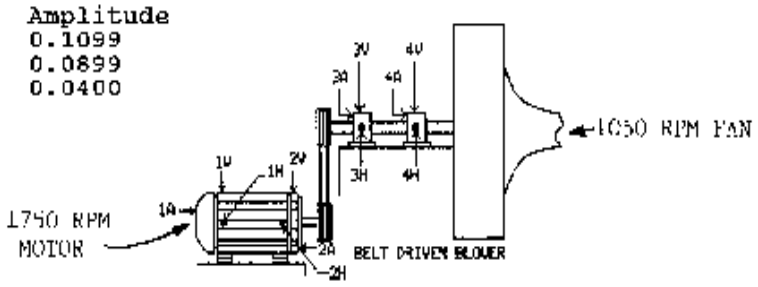
© Copyright 2001 Technical Associates of Charlotte, P.C.



2* x 2.000 = 0.0400 1 x RPM: 1750.00

Machine	Pos	Dir	Units	Date
304 AIRHAND 004	2	H	IN/S	Jan 24 1990 13:58

Frequency	1 x RPM	Amplitude
900	0.51	0.1099
1800	1.03	0.0899
3300	1.89	0.0400



T - Threshold adjust Q - Quit

FIGURE 1

© Copyright 2001 Technical Associates of Charlotte, P.C.

SUMMARY OF RESULTS AND RECOMMENDATIONS FOR MACHINES FOUND TO HAVE PROBLEMS

(SORTED FIRST BY PROBLEM PRIORITY; THEN BY PROBLEM SEVERITY WITHIN PRIORITY)

Page 1 of 4

AS OF: 1/21/89

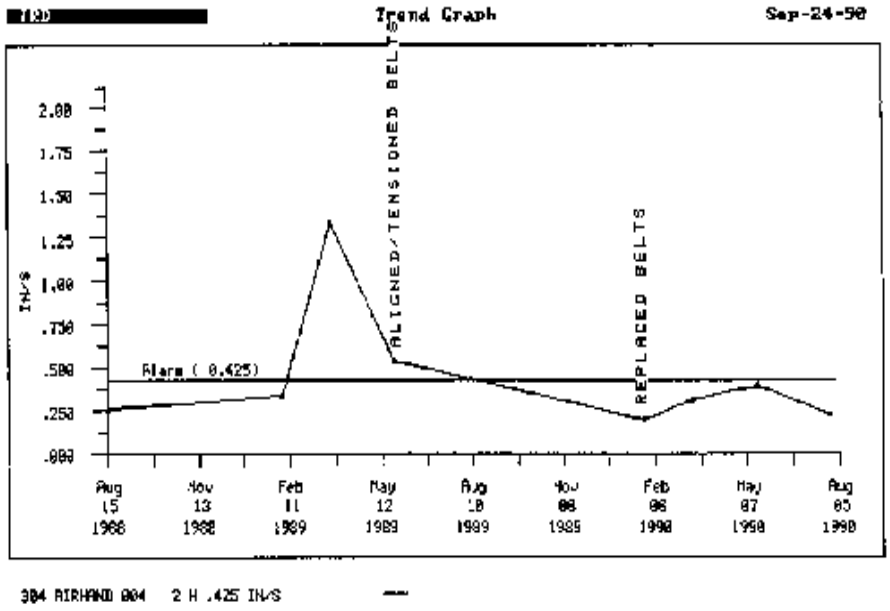
AREA	MACHINE COMPUTER NAME	INDICATED PROBLEM(S): UNBALANCE	MISALIGNMENT/SOFT FOOT	BENT SHAFT	LUBRICATION PROBLEM	BEARING PROBLEM	MECHAN. LOOSENESS	BELT/SHAVE PROBLEM	RESONANCE PROBLEM	GEAR PROBLEM	BLADE PASSING PROB.	CAVITATION	MOTOR ELECTRICAL	OIL WHIRL/WHIP	LOOSE FOUNDATION	VIB. MONITOR PROB.	SEAL/ROTOR RUB	OTHER (SEE REMARKS)	*PRIORITY OF PROBLEM	HIGHEST OVERALL VIBRATION: POSITION NO.	LEVEL: m = mils, l = in/sec, g = g's	RECOMMENDATIONS & REMARKS	REF PLOT NO.
	302 RCPCHIL 001	●																	①	3H	2.20	Extremely high amplitudes measured radially on the motor and chiller with the highest measured at position 3H (2.20 @ 1800 CPM) which indicates severe unbalance. Amplitudes have increased dramatically since September Survey as seen in Plot 37. Unbalance condition is probably caused by a balance weight being thrown off. Recommend shutting compressor off until unit is balanced.	36 37
	304 AIRHAND 004							●											2	2H	1.34	High amplitude measured at position 2H (.929 in/sec) at 1 times belt RPM measured on the motor indicates a belt/sheave problem. Check fan sheave for excessive runout and repair/replace as needed. Align and tension belts.	40
	324 AIRHAND 001	●																	2	2H	1.30	Motor unbalance still present at a slightly higher amplitude as seen in Plot 51. Balance motor as needed.	50 51
	863 RCPCOMP 001														●				2	2H	1.17	Possible loose or inadequate compressor foundation. Inspect compressor base for proper mounting that should include an inertia block or a vibration isolation system. Also, look for any loosened fasteners and tighten properly. Will continue to monitor closely.	89
	302 AIRHAND 001	●	●				●	●											2	4A	1.07	Belt misalignment and/or excessive sheave runout causing a high axial amplitude at position 1A combined with motor unbalance. Mechanical looseness is also indicated. Recommend checking sheaves for excessive runout and repair/replace as needed. Align and tension belts. Tighten all associated hold down bolts. Unbalance condition may be caused by sheave problem, therefore, trend this in future surveys.	33 34 35
	318 COOLTWR 001	●																	2	2H	.913	Motor and/or Drive Shaft Unbalance, likely combined with Drive Shaft Misalignment. Check drive shaft for looseness and runout and correct as necessary.	49

*① = TAKE IMMEDIATE REPAIR ACTION DUE TO SEVERE PROBLEM(S) NOTED (SEE "RECOMMENDATIONS/REMARKS" FOR THIS MACHINE)
2 = SCHEDULE REPAIRS TO CORRECT PROBLEMS NOTED AT CONVENIENCE (SEE "RECOMMENDATIONS/REMARKS" FOR THIS MACHINE)
3 = DESIGNATED PROBLEM DOES NOT WARRANT ACTION AT THIS TIME (CONTINUE ONLY WITH RELIABILITY SURVEY TREND ANALYSIS)

TECHNICAL ASSOCIATES OF CHARLOTTE, INC.
(704) 333-9011
© COPYRIGHT 1988

FIGURE 2

© Copyright 2001 Technical Associates of Charlotte, P.C.



IRD Mechanalysis Data Analysis System

Machine 304 AIRHAND 004 Pos 2 Dir H Units IN/S Alarm .425

Ampl	Insp Code	Date mm/dd/yy	Time	Ampl	Insp Code	Date mm/dd/yy	Time
.255	0	08-15-88	10:50				
.335	0	02-02-89	10:09				
1.34	0	03-22-89	16:36				
.537	0	05-23-89	16:54				
.195	0	01-24-90	13:58				
.313	0	03-13-90	14:23				
.396	0	05-15-90	18:00				
.223	0	07-26-90	15:38				
.223	0	07-26-90	15:38				

FIGURE 3

© Copyright 2001 Technical Associates of Charlotte, P.C.

EFFECT OF TIGHTENING LOOSE BELTS ON MOTOR VIBRATION

I. BACKGROUND

During a Predictive Maintenance Program data collection survey by Technical Associates' Analysts, a belt-driven air handler, 058 AIR HANDLER 002, was noted as having a loose set of belts and an unusual motor vibration of .380 in/sec at 2700 CPM. The .380 in/sec vibration violated the Band 4 alarm of .160 in/sec as seen in Plot 1. A machine diagram showing measurement locations is shown in the upper corner of the spectrum in Plot 1. The motor speed for this unit was nominally 1750 RPM and a hand-held tachometer showed the fan speed was 1530 RPM. A tachometer fan speed measurement is used to identify fan speed due to the possibility of belt slippage and sheave wheel size changes that could affect fan speed.

Since the 2700 CPM frequency was not related to either the motor or fan speeds (1X, 2X, 3X, etc.), another source needed to be considered. The motor and sheave wheel sizes were estimated, along with their center-to-center distance. Then an approximation of the belt frequency as being around 540 CPM was calculated. This identified the 2700 CPM and other frequencies in Plot 1 as harmonics of belt frequency. A Reliability Survey Report was issued reflecting this result. These reports are given after each Reliability Survey giving a continually updated evaluation of condition of each machine included in the plant Predictive Maintenance Program.

II. CONCLUSIONS

A. The high amplitude 2700 CPM frequency was found to be due to a 5X belt frequency vibration. The most likely cause of this high vibration as well as that by the other belt frequency harmonics was concluded to be loose belts.

III. RECOMMENDATIONS

A. Recommendations were made to tighten the loose belts to the proper tension.

B. It was also recommended the mechanic closely inspect both the motor and fan sheave for signs of wear which can contribute to belt looseness problems as well as accelerate wear of the belts themselves.

C. It was also recommended the mechanic closely align the motor sheave with respect to the fan sheave.

IV. RESULTS OF CORRECTIVE ACTIONS

After the belts were properly tightened, the 2700 CPM amplitude was reduced almost 80% from .380 in/sec to .078 in/sec as seen in Plot 2. These types of improvements also result in improved performance and reduced energy costs.

CF8 R-9210-1

© Copyright 2001 Technical Associates of Charlotte, P.C.

EFFECT OF TIGHTENING WORN BELTS ON MOTOR VIBRATION

SUMMARY OF RESULTS AND RECOMMENDATIONS FOR MACHINES FOUND TO HAVE PROBLEMS

(SORTED FIRST BY PROBLEM PRIORITY; THEN BY PROBLEM SEVERITY WITHIN PRIORITY)

Page 2 of 2

AS OF: 11/19/88

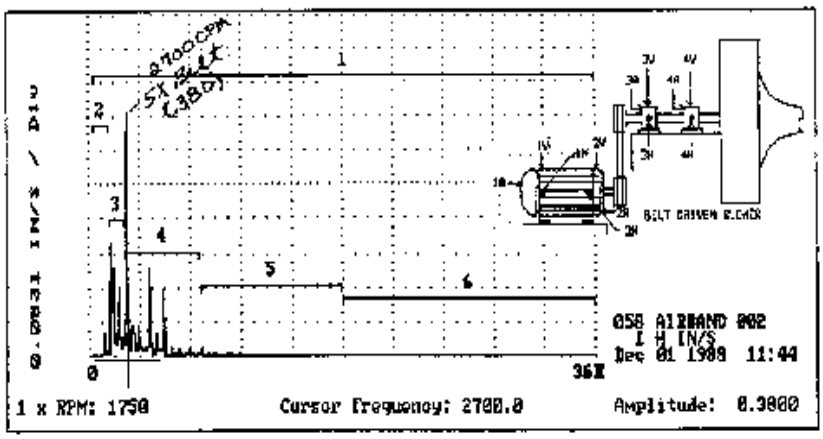
AREA	MACHINE COMPUTER NAME	INDICATED PROBLEM(S): UNBALANCE	MISALIGNMENT/SOFT FOOT	BENT SHAFT	LUBRICATION PROBLEM	BEARING PROBLEM	MECHAN. LOOSENESS	BELT/SHEAVE PROBLEM	RESONANCE PROBLEM	GEAR PROBLEM	BLADE PASSING PROB.	CAVITATION	MOTOR ELECTRICAL	OIL WHIRL/WHIP	LOOSE FOUNDATION	VIB. MONITOR PROB.	SEAL/ROTOR RUB	OTHER (SEE REMARKS)	*PRIORITY OF PROBLEM	HIGHEST OVERALL VIBRATION: POSITION NO.	LEVEL: m = mils, i = inches, g = g's	RECOMMENDATIONS & REMARKS	SEE PLOT NO.
	096 REPCOMP 001							●											3	1A	.191	Vibration at motor speed on both motor and compressor is likely caused by motor sheave eccentricity with only a slight chance of motor unbalance. Trend to determine if vibration is increasing.	35
	092 AIRHAND 003	●																●	3	2V	.578	Motor unbalance, possibly caused by sheave, results in moderately high vibration. Trend only.	69
	944 AIRHAND 003						●											●	3	3A	.556	A 2 X motor speed excitation at fan bearing caused a band 4 alarm of modest amplitude. Trend only.	77
	058 AIRHAND 002						●												3	1H	.549	A bell harmonic (5 X belt) caused a band 4 alarm at position 1H is likely caused by loose belts. Trend only at this time.	44
	058 AIRHAND 001	●																	3	2H	.479	Motor sheave unbalance. Trend only.	61

*(1) = TAKE IMMEDIATE REPAIR ACTION DUE TO SEVERE PROBLEM(S) NOTED (SEE "RECOMMENDATIONS/REMARKS" FOR THIS MACHINE)
2 = SCHEDULE REPAIRS TO CORRECT PROBLEMS NOTED AT CONVENIENCE. (SEE "RECOMMENDATIONS/REMARKS" FOR THIS MACHINE)
3 = DESIGNATED PROBLEM DOES NOT WARRANT ACTION AT THIS TIME. (CONTINUE ONLY WITH RELIABILITY SURVEY TREND ANALYSIS)

TECHNICAL ASSOCIATES OF CHARLOTTE, INC.
(704) 333-9011

FIGURE 1

© Copyright 2001 Technical Associates of Charlotte, P.C.



1% x 1.000 = 0.0100 1 x RPM: 1750.00

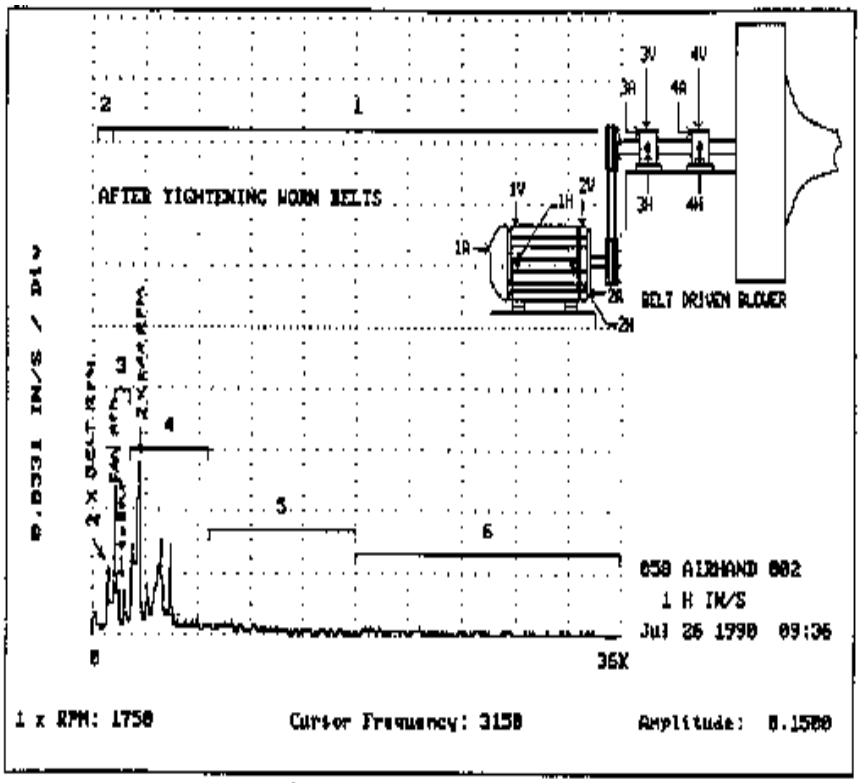
Machine	Pos	Dir	Units	Date
058 AIRHAND 002	1	H	IN/S	Dec 01 1988 11:44

Frequency	1 x RPM	Amplitude
1080.0	0.617	0.0349 – 2X BELT
1530.0	0.874	0.1751 – FAN
1800.0	1.029	0.1400 – MOTOR
2160.0	1.234	0.1099 – 4X BELT
2700.0	1.543	0.3800 – 5X BELT
3060.0	1.749	0.0500
3510.0	2.006	0.0450
4230.0	2.417	0.1400
4770.0	2.726	0.0349
5310.0	3.034	0.1050
5850.0	3.343	0.0149
6390.0	3.651	0.0100
7110.0	4.063	0.0100
8010.0	4.577	0.0149

T – Threshold adjust Q – Quit

PLOT 1

© Copyright 2001 Technical Associates of Charlotte, P.C.



2% x 0.500 = 0.0100 1 x RPM: 1750.00

Machine	Pos	Dir	Units	Date
058 AIRHAND 002	1	H	IN/S	Jul 26 1990 09:36

Frequency	1 x RPM	Amplitude	Frequency	1 x RPM	Amplitude
90	0.051	0.0250	9540	5.451	0.0125
1080	0.617	0.0575 = 2X BELT			
1530	0.874	0.1250 = FAN RPM			
1800	1.029	0.0375 = MOTOR RPM			
2160	1.234	0.0375 = 4X BELT			
2700	1.543	0.0775 = 5X BELT (WAS .380 in/sec)			
3150	1.800	0.1500 = 2X FAN			
3690	2.109	0.0425			
4680	2.674	0.0800			
5040	2.880	0.0200			
5310	3.034	0.0750			
5580	3.189	0.0150			
5850	3.343	0.0150			
6390	3.651	0.0100			
6840	3.909	0.0125			
7200	4.114	0.0150			
9000	5.143	0.0100			

T - Threshold adjust Q - Quit

PLOT 2

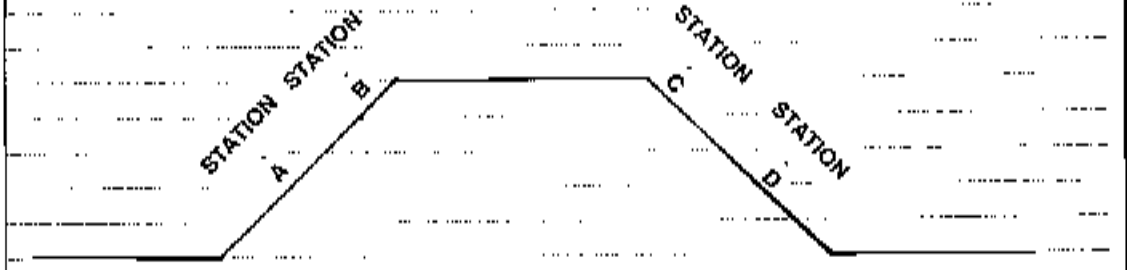
© Copyright 2001 Technical Associates of Charlotte, P.C.

Technical Associates of Charlotte, Inc.
347 N. Caswell Rd.
CHARLOTTE, NORTH CAROLINA 28204
(704) 333-9011

JOB
SHEET NO. OF 100
CALCULATED BY DATE
CHECKED BY DATE
SCALE

WALTER LIEBIG 2A,2B,2C, & 2D

EXCESSIVE MACHINE TOOL SPINDLE VIBRATION
DUE TO BELT FLUTTER



GENERAL ARRANGEMENT SHOWING 4 STATIONS

VIEW OF A SINGLE STATION OF FOUR

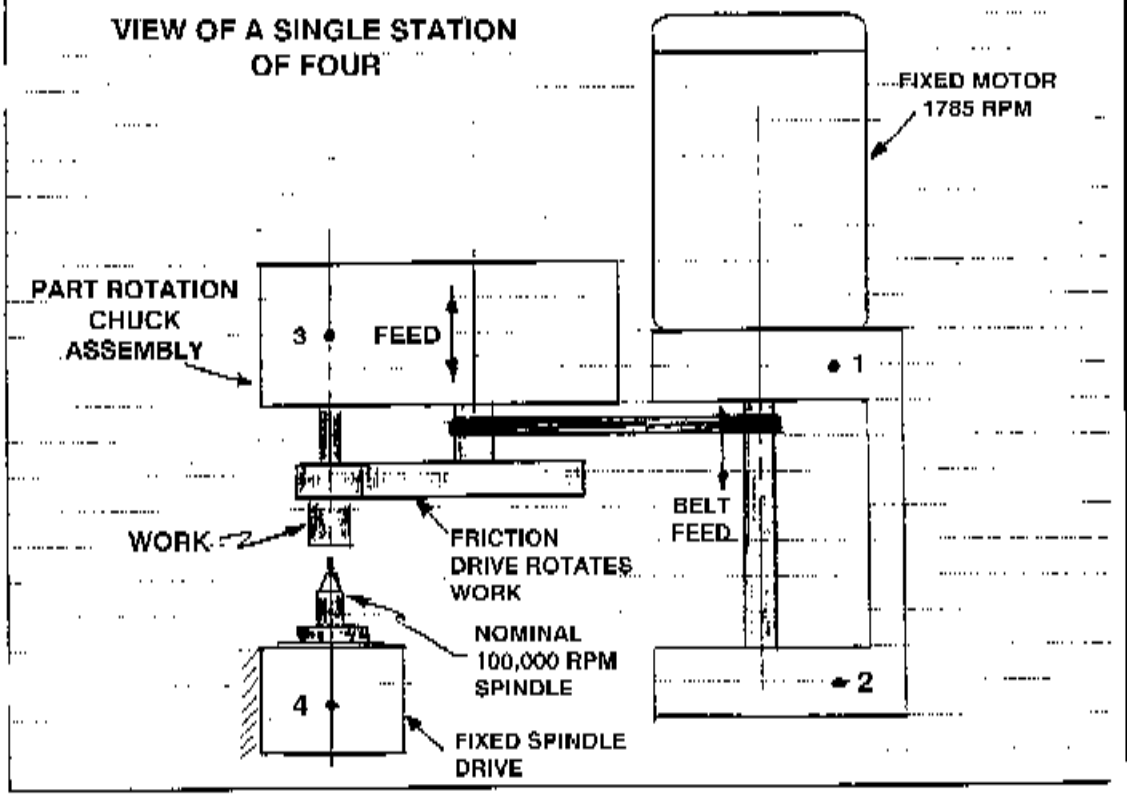


FIGURE 1

BMT-07

© Copyright 2001 Technical Associates of Charlotte, P.C.

SUMMARY OF RESULTS AND RECOMMENDATIONS FOR MACHINES FOUND TO HAVE PROBLEMS

(SORTED FIRST BY PROBLEM PRIORITY, THEN BY PROBLEM SEVERITY WITHIN PRIORITY)

Page 3 of 4 AS OF MAY 1, 1990

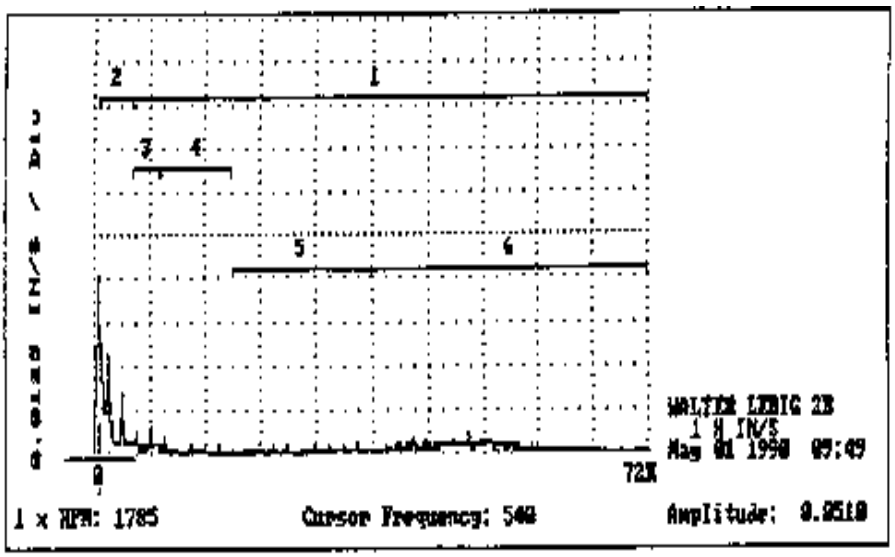
AREA	MACHINE COMPUTER NAME	UNBALANCE	MISALIGNMENT/SOFT FOOT	BENT SHAFT	LUBRICATION PROBLEM	BEARING PROBLEM	MECHAN. LOOSENESS	BELT/SHEAVE PROBLEM	RESONANCE PROBLEM	GEAR PROBLEM	BLADE PASSING PROB.	CAVITATION	MOTOR ELECTRICAL	OIL WHIRL/WHIP	LOOSE FOUNDATION	VIB. MONITOR PROB.	SEAL /ROTOR RUB	OTHER (SEE REMARKS)	*PRIORITY OF PROBLEM	POSITION NO.	LEVEL	RECOMMENDATIONS & REMARKS	SEE PLOT NO.
	EVA ALORE 2 HE										●								3	3H	.2271	Multiple harmonics of blade passing frequency (12,600) measured on the pump indicating a slight flow related problem. Recommend checking filters and associated piping for obstructions. Also confirm that gap between impeller vanes stationary diffuser vanes remains constant all the way around periphery.	28
	COND WTR VMUP22								●						●				3	4AA	.3501	Possible slight resonance due to loose foundation. Check foundation bolts for looseness. Trend in future surveys.	20
	WALTER LEBIG 2B							●											3	1H 2H	.1681 .1121	Belt flutter indicated at Positions 1H and 2H (see Plots 9 and 10. Belt was physically observed moving up and down on shaft, causing vibration to increase. Check belt drive and correct as required. Note difference in vibration on WALTER LEBIG 2B & 2D as seen in Plot 11 at the same Position 2H. Trend in future surveys.	9 10 11
	COOL TWR VMUP21										●				●				3	1A-H	.1381	Slightly high vibration at 9 times RPM (16050). Possible blade pass and/or loose foundation. Trend in future surveys. Also, please confirm the number of vanes on impeller and contact Technical Associates.	12
	4A YORK CHILLER					●													3	2H 2H	.790 g/SE .1051	Incipient wear of inboard motor bearing indicated by increasing level of spike energy which has now exceeded alarm. Closely watch in future surveys and check bearing for proper lubrication.	4
	SALEM SEAT 1225												●						3	3V 4A 7V	.0831 .0871 .0631	Slight motor electrical problem causing an amplitude of .083 and .087 in/sec at 7200 CPM measured at Positions 3V and 4A respectively, on Motor 12. Trend in future surveys. Also, due to the summation of overall vibration it is causing a Band 1 alarm. Trend in future surveys.	29 30 31

*(1) = IMMEDIATE REPAIR ACTION DUE TO SEVERE PROBLEMS NOTED (SEE "RECOMMENDATIONS/REMARKS" FOR THIS MACHINE)
2 = SCHEDULE REPAIRS TO CORRECT PROBLEMS NOTED AT CONVENIENCE (SEE "RECOMMENDATIONS/REMARKS" FOR THIS MACHINE)
3 = INDICATED PROBLEM DOES NOT WARRANT ACTION AT THIS TIME (CONTINUE ONLY WITH RELIABILITY SURVEY TREND ANALYSIS)

TECHNICAL ASSOCIATES OF CHARLOTTE, INC.
(704) 333-8011

© COPYRIGHT 1989

FIGURE 2



002% x 0.200 = 0.0040 1 x RPM: 1785.00

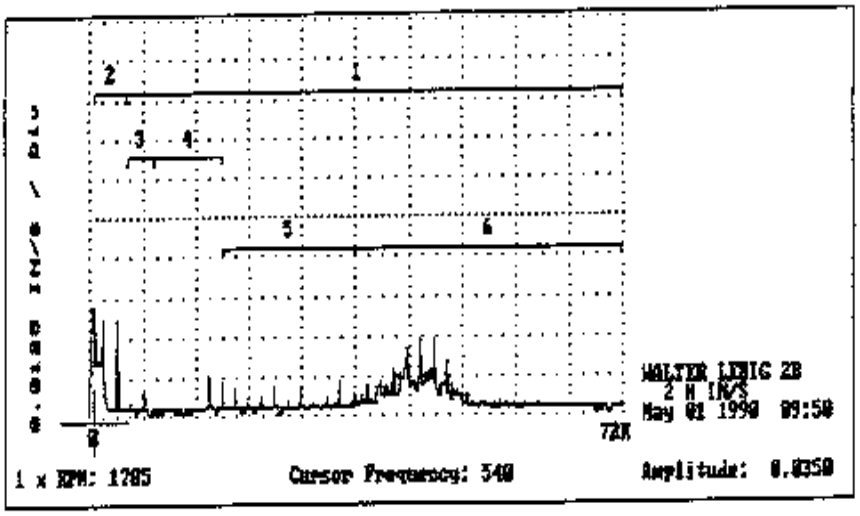
Machine	Pos	Dir	Units	Date
WALTER LEBIG 2B	1	H	IN/S	May 01 1990 09:4

Frequency	1 x RPM	Amplitude
540	0.30	0.0510
1800	1.01	0.0290
3600	2.02	0.0180
5400	3.03	0.0060
7200	4.03	0.0080
9000	5.04	0.0050
41220	23.09	0.0050
42840	24.00	0.0040
46440	26.02	0.0040
48240	27.03	0.0060
50040	28.03	0.0040
50760	28.44	0.0040

T - Threshold adjust Q - Quit

PLOT 1

© Copyright 2001 Technical Associates of Charlotte, P.C.



002% x 0.200 = 0.0040 1 x RPM: 1785.00

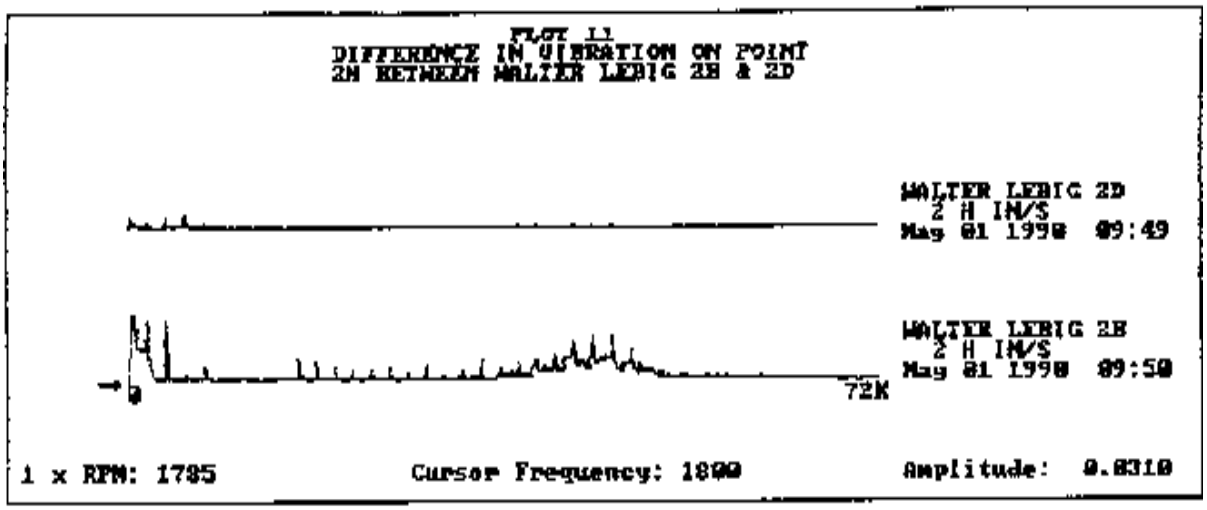
Machine	Pos	Dir	Units	Date
WALTER LEBIG 2B	2	H	IN/S	May 01 1990 09:5

Frequency	1 x RPM	Amplitude	Frequency	1 x RPM	Amplitude
540	0.30	0.0350	39420	22.08	0.0110
1080	0.61	0.0180	40140	22.49	0.0090
1800	1.01	0.0310	41040	22.99	0.0150
3600	2.02	0.0310	41940	23.50	0.0130
7200	4.03	0.0090	42840	24.00	0.0210
16020	8.97	0.0130	43380	24.30	0.0110
17820	9.98	0.0110	43740	24.50	0.0110
19620	10.99	0.0090	44640	25.01	0.0240
21420	12.00	0.0060	45360	25.41	0.0130
23220	13.01	0.0060	45900	25.71	0.0150
25020	14.02	0.0090	46440	26.02	0.0240
28620	16.03	0.0090	48240	27.03	0.0170
32220	18.05	0.0060	48960	27.43	0.0100
34020	19.06	0.0110	49680	27.83	0.0080
35820	20.07	0.0070	51120	28.64	0.0060
36900	20.67	0.0060			
37620	21.08	0.0100			

T - Threshold adjust Q - Quit

PLOT 2

© Copyright 2001 Technical Associates of Charlotte, P.C.



PLOT 3

© Copyright 2001 Technical Associates of Charlotte, P.C.

Glossary

REFERENCE: **Hewlett Packard DYNAMIC SIGNAL ANALYZER APPLICATIONS; "Effective Machinery Measurements Using Dynamic Signal Analyzers, Applications Notes 243-1; Oct. 1991.**

A

Acceleration. The time rate of change of velocity. Typical units are ft/s/s, meters/s/s, and G's (lG = 32.17 ft/s/s = 9.81 m/s/s). Acceleration measurements are usually made with accelerometers.

Accelerometer. Transducer whose output is directly proportional to acceleration. Most commonly use piezoelectric crystals to produce output.

Aliasing. A phenomenon which can occur whenever a signal is not sampled at greater than twice the maximum frequency component. Causes high frequency signals to appear at low frequencies. Aliasing is avoided by filtering out signals greater than 1/2 the sample rate.

Alignment. A condition whereby the axes of machine components are either coincident, parallel or perpendicular, according to design requirements.

Amplification Factor (Synchronous). A measure of the susceptibility of a rotor to vibration amplitude when rotational speed is equal to the rotor natural frequency (implies a flexible rotor). For imbalance type excitation, synchronous amplification factor is calculated by dividing the amplitude value at the resonant peak by the amplitude value at a speed well above resonance (as determined from a plot of synchronous response vs. rpm).

Amplitude. The magnitude of dynamic motion or vibration. Amplitude is expressed in terms of peak-to peak, zero-to-peak, or rms. For pure sine waves only, these are related as follows: rms = 0.707 times zero-to peak; peak-to-peak = 2 times zero-to-peak. DSAs generally read rms for spectral components, and peak for time domain components.

Anti-Aliasing Filter. A low-pass filter designed to filter out frequencies higher than 1/2 the sample rate in order to prevent aliasing.

Anti-Friction Bearing. See Rolling Element Bearing.

Asymetrical Support. Rotor support system that does not provide uniform restraint in all radial directions. This is typical for most heavy industrial machinery where stiffness in one plane may be substantially different than stiffness in the perpendicular plane. Occurs in bearings by design, or from preloads such as gravity or misalignment.

Asynchronous. Vibration components that are not related to rotating speed (also referred to as nonsynchronous).

Attitude Angle (Steady-State). The angle between the direction of steady-state preload through the bearing centerline, and a line drawn between the shaft centerline and the bearing centerline. (Applies to fluid film bearings.)

Auto Spectrum (Power Spectrum). DSA spectrum display whose magnitude represents the power at each frequency, and which has no phase. Rms averaging produces an auto spectrum.

Averaging. In a DSA, digitally averaging several measurements to improve accuracy or to reduce the level of asynchronous components. Refer to definitions of rms, time, and peak-hold averaging.

Axial. In the same direction as the shaft centerline.

Axial Position. The average position, or change in position, of a rotor in the axial direction with respect to some fixed reference position. Ideally the reference is a known position within the thrust bearing axial clearance or float zone, and the measurement is made with a displacement transducer observing the thrust collar.

© Copyright 2001 Technical Associates of Charlotte, P.C.

B

Balancing Resonance Speed(s). A rotative speed that corresponds to a natural resonance frequency.

Balanced Condition. For rotating machinery, a condition where the shaft geometric centerline coincides with the mass centerline.

Balancing. A procedure for adjusting the radial mass distribution of a rotor so that the mass centerline approaches the rotor geometric centerline.

Band-Pass Filter. A filter with a single transmission band extending from lower to upper cutoff frequencies. The width of the band is determined by the separation of frequencies at which amplitude is attenuated by 3 dB (0.707).

Bandwidth. The spacing between frequencies at which a band-pass filter attenuates the signal by 3 dB. In a DSA, measurement bandwidth is equal to [(frequency span)/ (number of filters) x (window factor)]. Window factors are: 1 for uniform , 1.5 for Hanning, and 3.63 for flat top.

Baseline Spectrum. A vibration spectrum taken when a machine is in good operating condition; used as a reference for monitoring and analysis.

Blade Passing Frequency. A potential vibration frequency on any bladed machine (turbine, axial compressor, fan, etc.). It is represented by the number of blades times shaft-rotating frequency.

Block Size. The number of samples used in a DSA to compute the Fast Fourier Transform. Also the number of samples in a DSA time display. Most DSAs use a block size of 1024. Smaller block size reduces resolution.

Bode. Rectangular coordinate plot of 1x component amplitude and phase (relative to a keyphasor) vs. running speed.

BPFO, BPFI. Common abbreviations for ball pass frequency of defects on outer and inner bearing races, respectively.

Bow. A shaft condition such that the geometric centerline of the shaft is not straight.

Brinneling (False). Impressions made by bearing rolling elements on the bearing race; typically caused by external vibration when the shaft is stationary.

C

Calibration. A test during which known values of the measured variable are applied to the transducer or readout instrument, and output readings varied or adjusted.

Campbell Diagram. A mathematically constructed diagram used to check for coincidence of vibration sources (i.e. 1 x imbalance, 2 x misalignment) with rotor natural resonances. The form of the diagram is a rectangular plot of resonant frequency (y-axis) vs excitation frequency (x-axis). Also known as an interference diagram.

Cascade Plot. See Spectral Map.

Cavitation. A condition which can occur in liquid handling machinery (e.g. centrifugal pumps) where a system pressure decrease in the suction line and pump inlet lowers fluid pressure and vaporization occurs. The result is mixed flow which may produce vibration.

Center Frequency. For a bandpass filter, the center of the transmission band.

Charge Amplifier. Amplifier used to convert accelerometer output impedance from high to low, making calibration much less dependent on cable capacitance.

© Copyright 2001 Technical Associates of Charlotte, P.C.

Coherence. The ratio of coherent output power between channels in a dual-channel DSA. An effective means of determining the similarity of vibration at two locations, giving insight into the possibility of cause and effect relationships.

Constant Bandwidth Filter. A band-pass filter whose bandwidth is independent of center frequency. The filters simulated digitally in a DSA are constant band width.

Constant Percentage Bandwidth. A band-pass filter whose bandwidth is a constant percentage of center frequency. 1/3 octave filters, including those synthesized in DSAs, are constant percentage bandwidth.

Critical Machinery. Machines which are critical to a major part of the plant process. These machines are usually unspared.

Critical Speeds. In general, any rotating speed which is associated with high vibration amplitude. Often, the rotor speeds which correspond to natural frequencies of the system.

Critical Speed Map. A rectangular plot of system natural frequency (y-axis) versus bearing or support stiffness (x-axis).

Cross Axis Sensitivity. A measure of off-axis response of velocity and acceleration transducers.

Cycle. One complete sequence of values of a periodic quantity.

D

Damping. The quality of a mechanical system that restrains the amplitude of motion with each successive cycle. Damping of shaft motion is provided by oil in bearings, seals, etc. The damping process converts mechanical energy to other forms, usually heat.

Damping, Critical. The smallest amount of damping required to return the system to its equilibrium position without oscillation.

Decibels (dB). A logarithmic representation of amplitude ratio, defined as 20 times the base ten logarithm of the ratio of the measured amplitude to a reference. DbV readings, for example, are referenced to 1 volt rms. Db amplitude scales are required to display the full dynamic range of a DSA.

Degrees Of Freedom. A phrase used in mechanical vibration to describe the complexity of the system. The number of degrees of freedom is the number of independent variables describing the state of a vibrating system.

Digital Filter. A filter which acts on data after it has been sampled and digitized. Often used in DSAs to provide anti-aliasing protection after internal re-sampling.

Differentiation. Representation in terms of time rate of change. For example, differentiating velocity yields acceleration. In a DSA, differentiation is performed by multiplication by *jw*, where *w* is frequency multiplied by 2þ. (Differentiation can also be used to convert displacement to velocity.)

Discrete Fourier Transform. A procedure for calculating discrete frequency components (filters or lines) from sampled time data. Since the frequency domain result is complex (i.e., real and imaginary components), the number of points is equal to half the number of samples.

Displacement. The change in distance or position of an object relative to a reference.

© Copyright 2001 Technical Associates of Charlotte, P.C.

Displacement Transducer. A transducer whose output is proportional to the distance between it and the measured object (usually the shaft).

DSA. See Dynamic Signal Analyzer.

Dual Probe. A transducer set consisting of displacement and velocity transducers. Combines measurement of shaft motion relative to the displacement transducer with velocity of the displacement transducer to produce absolute motion of the shaft.

Dual Voting. Concept where two independent inputs are required before action (usually machine shutdown is taken. Most often used with axial position measurements, where failure of a single transducer might lead to an unnecessary shutdown.

Dynamic Motion. Vibratory motion of a rotor system caused by mechanisms that are active only when the rotor is turning at speeds above slow roll speed.

Dynamic Signal Analyzer (DSA). Vibration analyzer that uses digital signal processing and the Fast Fourier Transform to display vibration frequency components. DSAs also display the time domain and phase spectrum, and can usually be interfaced to a computer.

E

Eccentricity, Mechanical. The variation of the outer diameter of a shaft surface when referenced to the true geometric centerline of the shaft. Out-of-roundness.

Eccentricity Ratio. The vector difference between the bearing centerline and the average steady-state journal centerline.

Eddy Current. Electrical current which is generated (and dissipated) in a conductive material in the presence of an electromagnetic field.

Electrical Runout. An error signal that occurs in eddy current displacement measurements when shaft surface conductivity varies.

Engineering Units. In a DSA, refers to units that are calibrated by the user (e.g., in/s, g's).

External Sampling. In a DSA, refers to control of data sampling by a multiplied tachometer signal. Provides a stationary display of vibration with changing speed.

F

Fast Fourier Transform (FFT). A computer (or microprocessor) procedure for calculating discrete frequency components from sampled time data. A special case of the discrete Fourier transform where the number of samples is constrained to a power of 2.

Filter. Electronic circuitry designed to pass or reject a specific frequency band.

Finite Element Modeling. A computer aided design technique for predicting the dynamic behavior of a mechanical system prior to construction. Modeling can be used, for example, to predict the natural frequencies of a flexible rotor.

Flat Top Filter. DSA window function which provides the best amplitude accuracy for measuring discrete frequency components.

Fluid-Film Bearing. A bearing which supports the shaft on a thin film of oil. The fluid-film layer may be generated by journal rotation (hydrodynamic bearing), or by externally applied pressure (hydrostatic bearing).

© Copyright 2001 Technical Associates of Charlotte, P.C.

Forced Vibration. The oscillation of a system under the action of a forcing function. Typically forced vibration occurs at the frequency of the exciting force.

Free Vibration. Vibration of a mechanical system following an initial force—typically at one or more natural frequencies.

Frequency. The repetition rate of a periodic event, usually expressed in cycles per second (Hz), revolutions per minute (rpm), or multiples of a rotational speed (orders). Orders are commonly referred to as 1x for rotational speed, 2x for twice rotational speed, etc.

Frequency Response. The amplitude and phase response characteristics of a system.

G

G. The value of acceleration produced by the force of gravity.

Gear Mesh Frequency. A potential vibration frequency on any machine that contains gears; equal to the number of teeth multiplied by the rotational frequency of the gear.

H

Hanning Window. DSA window function that provides better frequency resolution than the flat top window, but with reduced amplitude accuracy.

Harmonic. Frequency component at a frequency that is an integer multiple of the fundamental frequency.

Heavy Spot. The angular location of the imbalance vector at a specific lateral location on a shaft. The heavy spot typically does not change with rotational speed.

Hertz (Hz). The unit of frequency represented by cycles per second.

High Spot. The angular location on the shaft directly under the vibration transducer at the point of closest proximity. The high spot can move with changes in shaft dynamics (e.g., from changes in speed).

High-Pass Filter. A filter with a transmission band starting at a lower cutoff frequency and extending to (theoretically) infinite frequency.

Hysteresis. Non-uniqueness in the relationship between two variables as a parameter increases or decreases. Also called deadband, or that portion of a system's response where a change in input does not produce a change in output.

I

Imbalance. Unequal radial weight distribution on a rotor system; a shaft condition such that the mass and shaft geometric centerlines do not coincide.

Impact Test. Response test where the broad frequency range produced by an impact is used as the stimulus. Sometimes referred to as a bump test.

Impedance, Mechanical. The mechanical properties of a machine system (mass, stiffness, damping) that determine the response to periodic forcing functions.

Influence Coefficients. Mathematical coefficients that describe the influence of system loading on system deflection.

Integration. A process producing a result that, when differentiated, yields the original quantity. Integration of acceleration, for example, yields velocity. Integration is performed in a DSA by dividing by *jw*, where *w* is frequency multiplied by 2þ. (Integration is also used to convert velocity to displacement).

J

Journal. Specific portions of the shaft surface from which rotor applied loads are transmitted to bearing supports.

© Copyright 2001 Technical Associates of Charlotte, P.C.

K

Keyphasor. A signal used in rotating machinery measurements, generated by a transducer observing a once-per-revolution event. The keyphasor signal is used in phase measurements for analysis and balancing. (Keyphasor is a Bently Nevada trade name.)

L

Lateral Location. The definition of various points along the shaft axis of rotation.

Lateral Vibration. See Radial Vibration.

Leakage. In DSAs, a result of finite time record length that results in smearing of frequency components. Its effects are greatly reduced by the use of weighted window functions such as flat top and Hanning.

Linearity. The response characteristics of a linear system remain constant with input level. That is, if the response to input a is A, and the response to input b is B, then the response of a linear system to input (a + b) will be (A + B). An example of a non-linear system is one whose response is limited by mechanical stop, such as occurs when a bearing mount is loose.

Lines. Common term used to describe the filters of a DSA (e.g., 400 line analyzer).

Linear Averaging. See Time Averaging.

Low-Pass Filter. A filter whose transmission band extends from dc to an upper cutoff frequency.

M

Mechanical Runout. An error in measuring the position of the shaft centerline with a displacement probe that is caused by out-of-roundness and surface imperfections.

Micrometer (MICRON). One millionth (.000001) of a meter. (1 micron = 1 x E-6 meters = 0.04 mils.)

MIL. One thousandth (0.001) of an inch. (1 mil = 25.4 microns.)

Modal Analysis. The process of breaking complex vibration into its component modes of vibration, very much like frequency domain analysis breaks vibration down to component frequencies.

Mode Shape. The resultant deflected shape of a rotor at a specific rotational speed to an applied forcing function. A three-dimensional presentation of rotor lateral deflection along the shaft axis.

Modulation, Amplitude (AM). The process where the amplitude of a signal is varied as a function of the instantaneous value of another signal. The first signal is called the carrier, and the second signal is called the modulating signal. Amplitude modulation produces a component at the carrier frequency, with adjacent components (sidebands) at the frequency of the modulating signal.

Modulation, Frequency (FM). The process where the frequency of the carrier is determined by the amplitude of the modulating signal. Frequency modulation produces a component at the carrier frequency, with adjacent components (sidebands) at the frequency of the modulating signal.

N

Natural Frequency. The frequency of free vibration of a system. The frequency at which an undamped system with a single degree of freedom will oscillate upon momentary displacement from its rest position.

© Copyright 2001 Technical Associates of Charlotte, P.C.

Nodal Point. A point of minimum shaft deflection in a specific mode shape. May readily change location along the shaft axis due to changes in residual imbalance or other forcing function, or change in restraint such as increased bearing clearance.

Noise. Any component of a transducer output signal that does not represent the variable intended to be measured.

Nyquist Criterion. Requirement that a sampled system sample at a frequency greater than twice the highest frequency to be measured.

Nyquist Plot. A plot of real versus imaginary spectral components that is often used in servo analysis. Should not be confused with a polar plot of amplitude and phase of 1x vibration.

O

Octave. The interval between two frequencies with a ratio of 2 to 1.

Oil Whirl/Whip. An unstable free vibration whereby a fluid-film bearing has insufficient unit loading. Under this condition, the shaft centerline dynamic motion is usually circular in the direction of rotation. Oil whirl occurs at the oil flow velocity within the bearing, usually 40 to 49% of shaft speed. Oil whip occurs when the whirl frequency coincide with (and becomes locked to) a shaft resonant frequency. (Oil whirl and whip can occur in any case where fluid is between two cylindrical surfaces.)

Orbit. The path of the shaft centerline motion during rotation. The orbit is observed with an oscilloscope connected to x and y-axis displacement transducers. Some dual-channel DSAs also have the ability to display orbits.

Oscillator-Demodulator. A signal conditioning device that sends a radio frequency signal to an eddy-current displacement probe, demodulates the probe output, and provides output signals proportional to both the average and dynamic gap distances. (Also referred to as Proximitor, a Bently Nevada trade name.)

P

Peak Hold. In a DSA, a type of averaging that holds the peak signal level for each frequency component.

Period. The time required for a complete oscillation or for a single cycle of events. The reciprocal of frequency.

Phase. A measurement of the timing relationship between two signals, or between a specific vibration event and a keyphasor pulse.

Piezoelectric. Any material which provides a conversion between mechanical and electrical energy. For a piezoelectric crystal, if mechanical stresses are applied on two opposite faces, electrical charges appear on some other pair of faces.

Polar Plot. Polar coordinate representation of the locus of the 1x vector at a specific lateral shaft location with the shaft rotational speed as a parameter.

Power Spectrum. See Auto Spectrum.

Preload, Bearing. The dimensionless quantity that is typically expressed as a number from zero to one where a preload of zero indicates no bearing load upon the shaft, and one indicates the maximum preload (i.e., line contact between shaft and bearing).

Preload, External. Any of several mechanisms that can externally load a bearing. This includes ‘ soft” preloads such as process fluids or gravitational forces as well as “hard” preloads from gear contact forces, misalignment, rubs, etc.

© Copyright 2001 Technical Associates of Charlotte, P.C.

R

Radial. Direction perpendicular to the shaft centerline.

Radial Position. The average location, relative to the radial bearing centerline, of the shaft dynamic motion.

Radial Vibration. Shaft dynamic motion or casing vibration which is in a direction perpendicular to the shaft centerline.

Real-Time Analyzer. See Dynamic Signal Analyzer.

Real-Time Rate. For a DSA, the broadest frequency span at which data is sampled continuously. Real-time rate is mostly dependent on FFT processing speed.

Rectangular Window. See Uniform Window.

Relative Motion. Vibration measured relative to a chosen reference. Displacement transducers generally measure shaft motion relative to the transducer mounting.

Repeatability. The ability of a transducer or readout instrument to reproduce readings when the same input is applied repeatedly.

Resolution. The smallest change in stimulus that will produce a detectable change in the instrument output.

Resonance. The condition of vibration amplitude and phase change response caused by a corresponding system sensitivity to a particular forcing frequency. A resonance is typically identified by a substantial amplitude increase, and related phase shift.

Rolling Element Bearing. Bearing whose low friction qualities derive from rolling elements (balls or rollers), with little lubrication.

Root Mean Square (rms). Square root of the arithmetical average of a set of squared instantaneous values. DSAs perform rms averaging digitally on successive vibration spectra.

Rotor, Flexible. A rotor which operates close enough to, or beyond its first bending critical speed for dynamic effects to influence rotor deformations. Rotors which cannot be classified as rigid rotors are considered to be flexible rotors.

Rotor, Rigid. A rotor which operates substantially below its first bending critical speed. A rigid rotor can be brought into, and will remain in, a state of satisfactory balance at all operating speeds when balanced on any two arbitrarily selected correction planes.

RPM Spectral Map. A spectral map of vibration spectra versus rpm.

Runout Compensation. Electronic correction of a transducer output signal for the error resulting from slow roll runout.

S

Seismic. Refers to an inertially referenced measurement or a measurement relative to free space.

Seismic Transducer. A transducer that is mounted on the case or housing of a machine and measures casing vibration relative to free space. Accelerometers and velocity transducers are seismic.

Signal Conditioner. A device placed between a signal source and a readout instrument

© Copyright 2001 Technical Associates of Charlotte, P.C.

to change the signal. Examples: attenuators, preamplifiers, charge amplifiers.

Signature. Term usually applied to the vibration frequency spectrum which is distinctive and special to a machine or component, system or subsystem at a specific point in time, under specific machine operating conditions, etc. Used for historical comparison of mechanical condition over the operating life of the machine.

Slow Roll Speed. Low rotative speed at which dynamic motion effects from forces such as imbalance are negligible.

Spectral Map. A three-dimensional plot of the vibration amplitude spectrum versus another variable, usually time or rpm.

Spectrum Analyzer. An instrument which displays the frequency spectrum of an input signal.

Stiffness. The spring-like quality of mechanical and hydraulic elements to elasticity deform under load.

Strain. The physical deformation, deflection, or change in length resulting from stress (force per unit area).

Subharmonic. Sinusoidal quantity of a frequency that is an integral submultiple of a fundamental frequency.

Subsynchronous. Component(s) of a vibration signal which has a frequency less than shaft rotative frequency.

Synchronous Sampling. In a DSA, it refers to the control of the effective sampling rate of data; which includes the processes of external sampling and computed resampling used in order tracking.

Time Averaging. In a DSA, averaging of time records that results in reduction of asynchronous components.

T

Time Record. In a DSA, the sampled time data converted to the frequency domain by the FFT. Most DSAs use a time record of 1024 samples.

Torsional Vibration. Amplitude modulation of torque measured in degrees peak-to-peak referenced to the axis of shaft rotation.

Tracking Filter. A low-pass or band-pass filter which automatically tracks the input signal. A tracking filter is usually required for aliasing protection when data sampling is controlled externally.

Transducer. A device for translating the magnitude of one quantity into another quantity.

Transient Vibration. Temporarily sustained vibration of a mechanical system. It may consist of forced or free vibration or both. Typically this is associated with changes in machine operating condition such as speed, load, etc.

Transverse Sensitivity. See Cross-Axis Sensitivity.

Trigger. Any event which can be used as a timing reference. In a DSA, a trigger can be used to initiate a measurement.

U

Unbalance. See Imbalance.

Uniform Window. In a DSA, a window function with uniform weighting across the time record.

© Copyright 2001 Technical Associates of Charlotte, P.C.

This window does not protect against leakage, and should be used only with transient signals contained completely within the time record.

Vector. A quantity which has both magnitude and direction (phase).

Waterfall Plot. See Spectral Map.

© Copyright 2001 Technical Associates of Charlotte, P.C.